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U. S. S. *SALT LAKE CITY*.

CONSTRUCTION, DESCRIPTION, AND OFFICIAL PRELIMINARY ACCEPTANCE TRIAL DATA.

BY LIEUT. COMDR. J. K. ESLER, U. S. NAVY, MEMBER.

The U. S. S. *Salt Lake City*, officially designated as Scout Cruiser No. 25, was authorized by Congress in an act approved 18 December, 1924, which authorized the President of the United States to undertake, prior to July 1, 1927, the construction of eight scout cruisers; and provided "That in the event of an international conference for the limitation of naval armaments the President is hereby empowered, in his discretion, to suspend in whole or in part any or all alterations or construction authorized in this act."

Contract for construction of the U. S. S. *Salt Lake City* was concluded on July 9, 1926, with the William Cramp and Sons Ship and Engine Building Company at Philadelphia, Pa. The contract specified that the vessel should be in all respects complete and ready for delivery within 36 months of the date of the contract and provided for deductions from the contract price in the event of delay in delivery. The contract specifically covered procedure in the event of suspension of work by order of the government. A clause in the contract provided that the government might declare the contract forfeited in the event of failure or omission on the part of the contractor to go forward with the work and to make satisfactory progress toward completion of the vessel. The contract was not transferable on the part of the contractor.

Preliminary plans had been made and approved and a considerable amount of material and machinery ordered by the contractor when, on 11 April, 1927, the William Cramp and Sons Ship and Engine Building Company notified the Navy Department that it was unable to proceed with the work under the contract for construction of the U. S. S. *Salt Lake City* and also the contract to furnish machinery for the U. S. S. *Pensacola* and a test boiler of the type to be used on these vessels, had suspended work under both contracts, and expressed its willingness to be declared in default. The Navy Department accordingly on 11 April, 1927, declared both contracts forfeited and on 12 April, 1927, invited all qualified shipbuilders to submit proposals for completing the work covered by both contracts.

On 14 April, 1927, a contract was concluded with the American Brown Boveri Electric Corporation of Camden, N. J., to complete all work required for the fulfillment of both contracts.

The American Brown Boveri Electric Corporation arranged to employ many of the more important designers and workmen who had been engaged in the construction of the *Salt Lake City* at the Cramp Shipyard and work was continued at Camden with very little interruption due to the transfer of the contracts.

The Bureaus of Engineering and of Construction and Repair were informed on May 16, 1929, by the American Brown Boveri Electric Corporation of the formation of the Marine Engineering Corporation, with Mr. J. F. Metten as President. The contractors

for Scout Cruisers Number (25) *Salt Lake City,* (26) *Northampton,* (27) *Chester,* (30) *Houston,* and (31) *Augusta,* agreed that the Marine Engineering Corporation should complete plans and engineering for the American Brown Boveri Electric Corporation in addition to the design and engineering incident to construction of scout cruisers 26-27-30 and 31.

The Bureaus approved the centralized design and planning office primarily because of the advantage of standardization resulting from having the plans for all scout cruisers developed by the same personnel and in general identical for all ships. A procedure was developed and adopted whereby machinery plans for the *Salt Lake City* and *Pensacola* (CL25 and 24) were made to coincide as closely as practicable with plans for the *Northampton* (CL26) at Quincy, Mass., the *Chester* (CL27) at Camden, N. J., the *Houston* (CL30) and *Augusta* (CL31) at Newport News. The Government later, October 31, 1927, entered into a contract with the Marine Engineering Corporation for the furnishing of a duplicate set of all working drawings, together with material and working schedules, orders, lists, and instructions and all other information furnished by it to the builders of Scout Cruisers Nos. 26-27-30 and 31, for the use of the Government in the construction of Scout Cruiser No. 28 (*Louisville*) and No. 29 (*Chicago*) at the Navy Yards at Puget Sound, Washington, and Mare Island, California, respectively.

As a result of the foregoing as pertaining to the construction of the U. S. S. *Salt Lake City,* the machinery installations of all eight vessels of this group are practically identical and in a great measure interchangeable. Spare propellers and propeller shafting for any one vessel may be used on any other vessel of the class. Much of the auxiliary machinery, and spare parts of boilers and main engines are interchangeable. Late developments in Marine Engineering design have been incorporated into all vessels. The only marked difference in machinery layout of the eight cruisers lies in the boiler room and evaporator room arrangement. On the *Salt Lake City* and *Pensacola* boilers are arranged in two fire-rooms, four boilers placed two abreast and back to back in each fireroom. On the 26-31 class boilers are arranged in four fire-rooms, two boilers in each. The distilling plant is located on the

starboard side of the after fireroom on the *Salt Lake City* and *Pensacola,* whereas it is located outside of boiler and engineroom spaces on the 26-31 class.

Standardization of plans for machinery of all eight light cruisers under construction was responsible for considerable delay in plans for and work on the machinery of the *Salt Lake City* and *Pensacola.* Auxiliary machinery was in general ordered for all six ships (excluding light cruisers No. 28 (*Louisville*) and No. 29 (*Chicago*) which were being constructed by Navy Yards from the same plans). Finished plans for all eight ships were combined in one set of finished plans, notation being made as to applicability on each plan. One plan index was made up for all ships of the group.

An incident which occurred on the trials illustrates the advantage of interchangeability of machinery. The crosshead casting of dynamo air pump was broken and could not be repaired satisfactorily or a new casting made in time to proceed with the trials without delay. The corresponding crosshead from one of the dynamo air pumps on the *Northampton* was obtained by an all-night automobile ride to Quincy, Mass., and return to Rockland, Me., and was found to fit perfectly.

Contracts for the U. S. S. *Salt Lake City* provided for guarantees as to oil consumption at various shaft horsepowers of the main engines, as well as endurance trials at full power and various trials to test maneuverability. Trials were to be conducted by the contractor and witnessed by government representatives. A bonus and penalty clause was contained in the contract to insure that contract weight be not exceeded because of treaty limitations. The bonus allowed was placed at $50,000. The contract displacement at the time of official preliminary acceptance trials was 7295.7 tons, which was 630.3 tons less than the 7926 tons allowed by the contract and the maximum bonus was allowed. Actual displacement at the time of standardization runs was 10,528.5 tons, this including fuel oil and some water ballast to compensate for stores.

The economy measured in pounds of fuel oil per knot was better than the contract guarantees on all fuel consumption trials except at fifteen knots where the contract guarantee was slightly exceeded. Under the terms of the contract no bonus was allowed.

The *Salt Lake City* is in general appearance noticeably larger and more formidable than earlier light cruisers. This impression is increased by the larger enclosed gun mounts, the two tripod masts, the two stacks and the flush deck. The teak deck extending unbroken from bow to stern is not as much taken up with boat stowage, etc., as on the earlier light cruisers and all resemblance to an overgrown destroyer is entirely lacking. The first impression on going on board is that of a large ship. The after deck is comparable with the quarter deck of the armored cruisers in space.

The wardroom, located on the main deck forward is about twice the size of the wardrooms of earlier light cruisers, and is fitted with adequate air ports. Officers' quarters are much the same as on earlier ships, in general, but the quarters for the crew are very much of an improvement over the crew's quarters on the other light cruisers.

General characteristics are as follows:

PRINCIPAL DIMENSIONS.

Length between perpendiculars, feet and inches	570	0
Length overall, feet and inches	585	8
Beam extreme, feet and inches	65	3
Draft to designer's W. L. (mean), feet and inches	19	6½
Corresponding displacement, tons	11,540	

TANK CAPACITIES.

Reserve feed water, normal, tons	133
Reserve feed water, including emergency tanks, tons	484
Ship's fresh water tanks (2), gallons each	6691

Reserve feed bottoms are located under the firerooms, the eight double bottom tanks adjacent to the keel being normally used and those immediately outboard of these being intended for emergency use.

Fuel oil tanks are located in double bottoms forward of the firerooms, amidships tanks extending up to the 2nd platform deck and wing tanks to 1st platform deck, also under and abaft the enginerooms, and outboard of firerooms. The tanks adjacent to emergency reserve feed water tanks are intended for emergency use only while tanks outboard of these are intended for regular

use. The lack of cofferdams between emergency fuel oil and reserve feed water tanks restricts the use of emergency reserve feed water tanks and in effect converts these tanks into cofferdams. As a result the carrying capacity as regards reserve feed water is below normal for this type of ship. It is believed, however, that with the excellent evaporating plant provided there will be little or no trouble in this respect.

MAIN ENGINES.

Four—Main turbines, Parsons, geared; one H. P., L. P. and cruising turbine operating through single reduction gears on each shaft. Astern turbines—all impulse blading—are located in the L. P. turbine casing. Turbine design in general follows the design on the *Richmond* class light cruisers, the principal departure being a cruising turbine for each unit. Each unit is designed to develop continuously 26,750 shaft horsepower. The cruising turbine is designed for operation through single reduction gearing and Metten hydraulic clutch to H. P. turbine shaft at speeds up to about twenty knots and to operate disconnected from main turbines at all times if desired, in which case it acts merely to drive the connected pumps. When operating independently the cruising turbine is governor controlled at a speed corresponding to about fifteen knots. Automatic operation of exhaust valves is provided for shifting cruising turbine exhaust from condenser to H. P. turbine when coupling, and from H. P. turbine to main condenser when uncoupling. Provision is made for bleeding steam from appropriate stages of the H. P. turbine to the auxiliary exhaust for use in feed water heaters and evaporators, and for using auxiliary exhaust in turbines if in excess. The number of electric motor driven auxiliaries makes it improbable that there will be any excess auxiliary exhaust steam under any condition of cruising. Each cruising turbine is fitted to drive main air pump, lubricating oil pump, turbine drain pump and one additional pump originally designed for use as a cruising feed pump, found unsuited for that purpose and retained for use as a make up feed pump on one unit and as a bilge pump on the other unit in each engine room. Gland steam exhauster system is installed on main and cruising turbines.

Direct connected pumps are as follows:

Main air pump—Edwards type.

Lubricating oil pump—Quinby Screw type.

Turbine drain pump—Special type of solid plunger single acting pump.

Make up feed pump or attached bilge pump—Reciprocating single acting.

Independent lubricating oil pumps are provided for use in case of the failure of attached pumps.

At cruising speeds the main air pump suction lines in each engine room may be cross connected, this permitting the operation of but one cruising turbine under this condition if desired, and also providing that one cruising turbine air pump may take condensate from both condensers in case of the failure of one cruising turbine unit.

The after engine room is the control engine room.

BOILERS.

8—Babcock and Wilcox-White Forster, Oil burning, arranged in two firerooms.

Furnace volume—one boiler—1137 cubic feet.

Heating surface—one boiler—11,880 square feet.

No. burners—13—per boiler—Cuyama type.

Steam Pressure—working—300 pounds gauge.

Authorized safety valve setting 310 pounds.

Steam—not superheated.

Soot blowers—Diamond type.

Periscopic smoke indicators—2 per boiler.

Burner telegraph—operated from after engine room.

Boiler safety valves—two duplex 4-inch on each boiler, Form E Consolidated Ashcroft Hancock Company, modified to include new design features.

PROPELLERS.

4—Solid, 3 blade, Manganese bronze.

Diameter—12 feet 0 inches.

Pitch—mean—11 feet 9 inches.

Heliocoidal area—72 square feet.

Projected area—62.2 square feet.

Designed for 366 R. P. M. at 107,000 S. H. P.

STEERING GEAR.

1—Electro-hydraulic screw gear type. Two separate power units each consisting of a 100 horsepower electric motor driving through spur gearing, of 1 to 4 ratio, a Waterbury pump, size 150, together with necessary control units. The rudder is actuated by a screw gear of the usual type but with the driven sleeves actuating the crosshead direct through pins in sliding blocks mounted in transverse slots in the crosshead. The screw shaft is fitted with a large spur gear on its forward end with a jaw pinion on each side constantly in mesh with the gear. Each of these pinions is driven direct by a size 150 Type K Waterbury oil motor with jaw clutch engagement.

FORCED DRAFT BLOWERS.

12—6 in each fireroom. Sturtevant vertical turbine driven—50,000 cubic feet per minute capacity each.

Intake trunks fitted with spring balanced automatic shutters of new design.

Arrangement is made for utilizing the motor driven engine room ventilating blowers for port use and for cruising speeds. The after fireroom has two intakes fitted with doors for control of opening by reach rod to passage on 2nd deck as well as from fireroom, thus regulating the amount of air diverted to the firerooms and permitting closing when necessary to start main blowers. The forward fireroom has only one such intake, supplied from forward engine room ventilating blowers. Independent ventilating system is installed for evaporator room for use when air pressure in after fireroom permits.

MAIN GENERATORS.

4—General Electric three wire 250 kilowatt turbine driven through reduction gears. Two sets installed in each engine room. Turbines are designed to operate condensing only. One set is fitted with exhaust connection to auxiliary exhaust line for emergency use or when in dry dock. Turbine speed 10,012 R.P.M., generator speed 1200 R.P.M.

The two sets in each engineroom are located directly above a dynamo condenser, which serves also as an auxiliary condenser. Dynamo air and circulating pumps are motor driven, the air pump being reciprocating and the circulating pump centrifugal. Connections are provided for draining condenser and air pump to reserve feed tanks and also for use of dynamo air pumps in drying out main and cruising turbines through turbine drains.

MAIN CONDENSERS.

4—One for each main propelling unit. Uniflow type with scoop injection and turbine driven direct connected vertical main circulating pumps for use when stopped, backing, or maneuvering at low speeds. The throttle for these pumps is located at the working platform and pump turbine exhaust valve is fitted with steam piston operated disk to open the exhaust valve automatically when pump throttle is opened. Main condenser cooling surface 12,692 square feet each. Absolute pressure gauges (mercury) are fitted on each main condenser in addition to the dial type spring actuated gauges. Augmentors and augmentor condensers are installed.

MAIN FEED PUMPS.

4—Worthington—turbine driven single stage—centrifugal, designed for 750 G.P.M. at 5200 R.P.M. with 400 pounds outlet pressure. Two pumps in each engineroom.

EMERGENCY FEED PUMPS.

2—Worthington vertical duplex, double acting, 16 1/2 inches × 12 inches × 24 inches—one pump in each fireroom. Two additional feed pumps—Worthington vertical duplex, double acting, 14 inches × 9 inches × 24 inches—are to be installed, one in each fire room, and connected to suction and discharge in the same manner as the emergency feed pumps. These pumps are in lieu of the cruising feed pumps originally required by specifications and it is contemplated will be used at cruising speeds instead of the turbine driven main feed pumps.

FUEL OIL BOOSTER AND RECIRCULATING PUMPS.

4—Worthington, reciprocating 14 inches × 11 inches × 18 inches—Two pumps in each fireroom.

FUEL OIL SERVICE PUMPS—RECIPROCATING.

8—Worthington, reciprocating, 5 1/2 inches × 4 inches × 8 inches duplex double acting. Four pumps in each fireroom.

PORT FUEL OIL SERVICE PUMPS.

2—Waterbury oil pumps motor driven. Capacity for speeds up to about 15 knots when using one fireroom.

FUEL OIL RECIRCULATING SYSTEM.

The fuel oil double bottoms under the engine rooms are provided with heating coils. Fuel oil in other stowage tanks is heated by means of the recirculating system. Two recirculating systems are provided, one forward and one aft, each capable of heating 250 G.P.M. of oil at 700 seconds Saybolt Furol from a temperature of 100 degrees to 170 degrees F.

DISTILLING PLANT.

2—Double effect L.P. submerged tube evaporator sets, rated capacity, 40,000 gallons per day.
2—Distiller condensers—horizontal type.
2—Tube nest drain heaters.
2—Vapor feed heaters.
2—Condensate coolers.
2—Turbine driven combination pumps—each with
1—reciprocating air pump 11 inches × 10 inches.
1—single suction volute distiller circulating pump, 400 G.P.M.
1—single cylinder brine overboard pump—single acting.
1—single cylinder fresh water pump—single acting.
1—Sturtevant type A-17 turbine 1300 R.P.M.
1—Evaporator feed pump motor driven—90 G.P.M.
1—Meter—2 measuring tanks.

It is believed that except in emergency one set will supply all fresh water required. Due to the number of electric driven auxiliaries it is believed that it may be necessary to bleed live steam at times to obtain sufficient pressure to operate one set.

REFRIGERATING PLANT.

2—Electric motor driven Kroeschell 2 ton Carbon dioxide compressors.
1—Carbon dioxide condenser.
1—Brine cooler.
2—Brine circulating pumps, centrifugal, motor driven.
2—Condenser circulating pumps, centrifugal, motor driven.
1—Ice making box, rated at 400 pounds per day.

It is believed that one machine will be sufficient for normal load. The Bureau of Construction and Repair have cognizance over small individual refrigerating machines installed in Officer's and Chief Petty Officer's pantries. These are of the General Electric Type.

FIRE AND FLUSHING PUMPS.

2—Worthington centrifugal, provided with motor drive at one end of impeller shaft for use on flushing system and direct connected turbine drive on other end of impeller shaft for use as fire pumps. One pump is located in each engine room.

MACHINE TOOLS.

1—Gap lathe 16 inches × 32 inches × 8 feet extension bed.
1—Engine lathe 16 inches × 10 feet.
1—Universal Milling Machine 28 inches × 10 inches × 18 inches.
1—Upright drill press—28 inches with sliding head.
1—Shaper 16 inches.
1—Sensitive drill 16 inches floor type.
1—Combined wet and dry wheel emery grinder.
1—Power hack saw.

All machine tools are fitted with independent motor drive.

SEARCHLIGHTS.

4—36-inch High Intensity.
2—24-inch Light weight high intensity, signal.
4—1000 Watt Incandescent, signal.

GENERAL ANNOUNCING SYSTEM.

General announcing system is installed with individual loud speakers in all parts of the ship where required.

STORAGE BATTERIES.

Forward battery room.

Auxiliary lighting battery 42 trays—230 Volt.
Battle telephone battery—8 trays—24 volts.
Automatic telephone battery—6 trays—12 volt.

After battery room.

Auxiliary battery—230 volts.

The forward batteries supply emergency power to the Fire control, Interior communication, radio, gyro, and forward auxiliary lighting circuits. The after battery supplies emergency power for steering gear motors, after gyro, after auxiliary lighting.

Dynamo condenser pump motors and port fuel oil service pump motors are connected to use battery in emergency.

GYRO COMPASS EQUIPMENT.

2—Gyro compasses Arma Mark IV are installed, one forward and one aft, with connections for use of forward and after storage batteries.

AERONAUTICAL INSTALLATION.

2—Catapults.
4—Planes.
1—Fabricated boom located amidships and extending about eight feet over either side, fitted with electric motor, operating three Waterbury oil motors for hoisting, topping, and training.

ANCHOR GEAR.

Motor driven windlass, with control on the forecastle.

BOAT STOWAGE.

Boats are stowed aft just forward of the mainmast. Boat handling booms can be controlled either from the starboard side or from the port side having electric motor and Waterbury oil motors for hoisting, topping and training.

BUILDER'S TRIALS.

Dock trials were held 28 August, 1929, and machinery appeared to be in general satisfactory. The first builder's trial was held 31 August, 1929. The vessel proceeded down the Delaware and operated at various speeds to try out the machinery and the ship, and to train the operating personnel preparatory to official preliminary acceptance trials. On this trial the main air pumps, driven by an extension shaft and worm gear from the cruising turbine pinion, pounded considerably and one finally sheared the pins provided in the connecting coupling for that purpose in case of overload. Main feed pumps of the centrifugal single stage type also gave trouble and waterhammers in the feed lines caused many leaking flange joints. The ship returned to the shipyard and alterations were undertaken to correct the defects noted.

On 20 October, 1929, the ship again went down to Delaware Bay for a second builder's trial. Main air pumps and main feed pumps were still unsatisfactory. It was determined also that feed tank capacity, the minimum allowed under the specifications, was insufficient for reliable operation at high speeds while maneuvering, due to the fact that the draft of the ship increased aft at high speeds and the resulting change in trim was sufficient to cause the water from the forward main feed tank to flow to the after main feed tank and cause it to overflow.

The Bureaus of Engineering and Construction and Repair, at the request of the contractor authorized a change under the contract to install two additional supplementary feed tanks in the after engine room, with equalizing connections to the after main feed tank, as it was considered desirable that these tanks be installed prior to the official trials. A modified type of shear pin coupling installed in the air pump drive, proved unsatisfactory and decision was made to install a friction coupling. Work continued on main feed pumps and the trouble was remedied by omission of vanes

in diffuser ring through which water was discharged from the impeller. Difficulty had been experienced on both builder's trials with the reciprocating fuel oil service pumps and considerable work was done in overhauling these. The cruising feed pumps, driven from the main air pump combination drive, proved entirely unsatisfactory for use as feed pumps due to the high pressure of the discharge and the connections to these pumps were broken and they were piped for use as make up feed pumps on the official trials. Air chambers and baffles were installed on main air pump discharges to cushion the discharge. An overflow was installed on main feed tanks to protect the tanks and piping from excessive pressure while operating.

The date of leaving the shipyard for the measured mile course at Rockland to hold official preliminary acceptance trials was set as 9 November, 1929. Preparations on the part of the shipbuilder had been most thorough as noted by the Trial Board and by representatives of the various Bureaus witnessing the trials.

Official preliminary acceptance trials were conducted by Mr. E. I. Cornbrooks, Vice President and General Manager of the New York Ship Building Company, assisted by officials and employees of the Company.

Rear Admiral S. E. W. Kittelle, U. S. Navy, was President of the Trial Board. He was assisted by other members of the Trial Board as follows:

Captain H. T. Wright (CC), U. S. N.; Lieutenant Commander W. K. Harrill, U. S. N.; Commander C. A. Bonvillian, U. S. N.; Lieutenant Commander J. W. Gregory, U. S. N.; Commander J. H. S. Dessez, U. S. N.

The trials were witnessed by the Inspector of Machinery and Assistant Superintending Constructor, representatives of the office of the Chief of Naval Operations, the Bureau of Engineering, the Bureau of Construction and Repair, the prospective Commanding, Executive, and Engineer Officers with other officers and enlisted men assigned to duty on board after commissioning, as well as the prospective Commanding Officer and Engineer Officer of the *Pensacola,* the sister ship nearing completion at the Navy Yard, New York.

Officials from the Bethlehem Shipbuilding Corporation and the Newport News Shipbuilding and Dry Dock Company and repre-

sentatives of many sub-contractors who had furnished boilers and auxiliary machinery for the ship were present to observe and assist in conducting the trials.

Standardization trials were started on Tuesday, 12 November, 1929. The weather was clear with a light haze, barometer 30.20, light variable airs, sea calm. During the standardization a total of twenty-four runs were made over the measured mile course, three runs at 20 knots, three at 25 knots, three at 28 knots, three at 30 knots, three at 32 knots, and eight at full speed ahead and one at full speed astern.

Standardization runs were continued on the second day, Wednesday, 13 November, 1929. The weather was overcast with light variable airs, barometer 30.28, sea smooth. Twelve runs were made, three at ten knots, three at 14 knots, three at 17 knots, and three at 24 3/4 knots. The last runs were made to check the 25 knot group obtained the previous day.

Following is a summary of R.P.M., S.H.P., and speed obtained on standardization runs:

AVERAGES OF RUNS MADE AT EACH SPEED.

R.P.M.	*Speed*	*S.H.P.*
90.69	10.22	1374
129.65	14.36	4695
158.54	17.47	8651
188.70	20.60	14,954
232.73	24.79	28,508
236.94	25.04	29,964
268.01	28.23	44,659
299.21	30.20	63,686
355.34	32.22	98,833
361.96	32.43	101,797
372.06	32.78	107,746

In general the machinery operated very satisfactorily, the exception being the fuel oil service pumps. Main feed pump speed regulators were blocked to prevent surging when pumps were operated in parallel. Boilers, forced draft blowers, and main propelling machinery were very satisfactory. In view of the limited opportunity which had been afforded for training the operating person-

nel prior to the standardization trials, the efficiency with which the trials were conducted was very commendable and reflects credit on the leading men and others of the trial crew.

On completion of standardization trials on the second day the ship proceeded to sea working up to twenty-five knots. At noon the four hour twenty-five knot fuel economy run was started and successfully completed at 4 P. M. During this run the weather was overcast, with light variable airs, sea smooth, barometer 30.28.

On the third day the two hour trials on main engines (without cruising turbines) at 20 knots, 15 knots, and at 10 knots were successfully completed, and a trial was held to demonstrate the facility with which cruising turbines could be coupled and uncoupled at various speeds.

On the fourth day, Friday, 15 November, 1929, the ship proceeded to sea but because of unfavorable weather, fog with occasional rain squalls, returned to anchorage at Rockland.

The weather was overcast with general northwest breezes, sea smooth, barometer 30.06 on Saturday, 16 November, and the vessel proceeded to sea to conduct the four hour 30 knot fuel economy trial. The wind gradually increased to force five and the sea became choppy. The trial was started at 8 A. M. and completed at noon. At 2 P. M. the fifteen knot fuel economy trial was started with cruising turbines connected and was completed successfully at 6 P. M.

On Sunday, 17 November, the ship conducted the four hour fuel consumption trial at 107,000 H. P., weather clear and sea smooth. This trial was begun at 9 A. M. and successfully completed at 1 P. M., after which the ship was continued at full power for two hours. The terms of the contract left the decision as to the maximum power developed on this trial to the discretion of the contractors in view of their responsibility for the conduct of the trials. The maximum power developed was 110,254 S.H.P. during the last half hour of the trial. After machinery has been run in and with a trained engineer's force there appears to be no question but that this power and the speed attained, 32.77 knots can be exceeded without difficulty. It is worthy of note that two of the six forced draft blowers in each of the two firerooms were

not in use during this trial and that the others were not operating at the maximum speed permitted.

On the seventh day the *Salt Lake City* proceeded to Vineyard Haven via Pollock Rip Slue and Nantucket Sound. The weather being unfavorable and the fog dense, no trials were conducted.

On the eighth day, Tuesday, 19 November, the weather was clear, sea moderate, the wind gradually increasing to about half a gale at one P. M. and continuing throughout the day. The four hour twenty knot fuel economy trial was started at 10.00 A. M. and satisfactorily completed at 2 P. M. The course was against the wind and sea during the first two hours and with the wind and sea during the last two hours. The effect of wind and sea are apparent in the following table of half hourly determinations of R.P.M. and S.H.P.

Time	*R.P.M.*	*S.H.P.*	
10:30	183.29	14,523	
11:00	183.13	14,670	
11:30	183.16	14,671	
12:00	183.09	14,356	
12:30	184.02	14,167	
1:00	183.72	13,421	
1:30	183.03	13,215	
2:00	183.14	13,583	
Average	183.32	14,077	Speed 20.01 knots.

The wind and sea having moderated the four hour ten knot fuel economy trial was started at 7 P. M. and successfully completed at 11:00 P. M.

The contractors did not desire to conduct the reversal tests and tests of steering gear at a greater speed than 30 knots because of uncertainty as to the effectiveness of turbine drainage systems, on which they contemplated making some changes before the delivery of the ship. At their request the trials at full power were postponed until the final acceptance trials, and preliminary reversal and steering tests conducted at 300 R.P.M. ahead and 200 R.P.M. astern. The elapsed time from giving the full speed astern signal when making 300 R.P.M. ahead (approx. 30 knots), until the ship was dead in the water was 3 minutes 15 seconds. The head reach was approximately 700 yards. With the ship

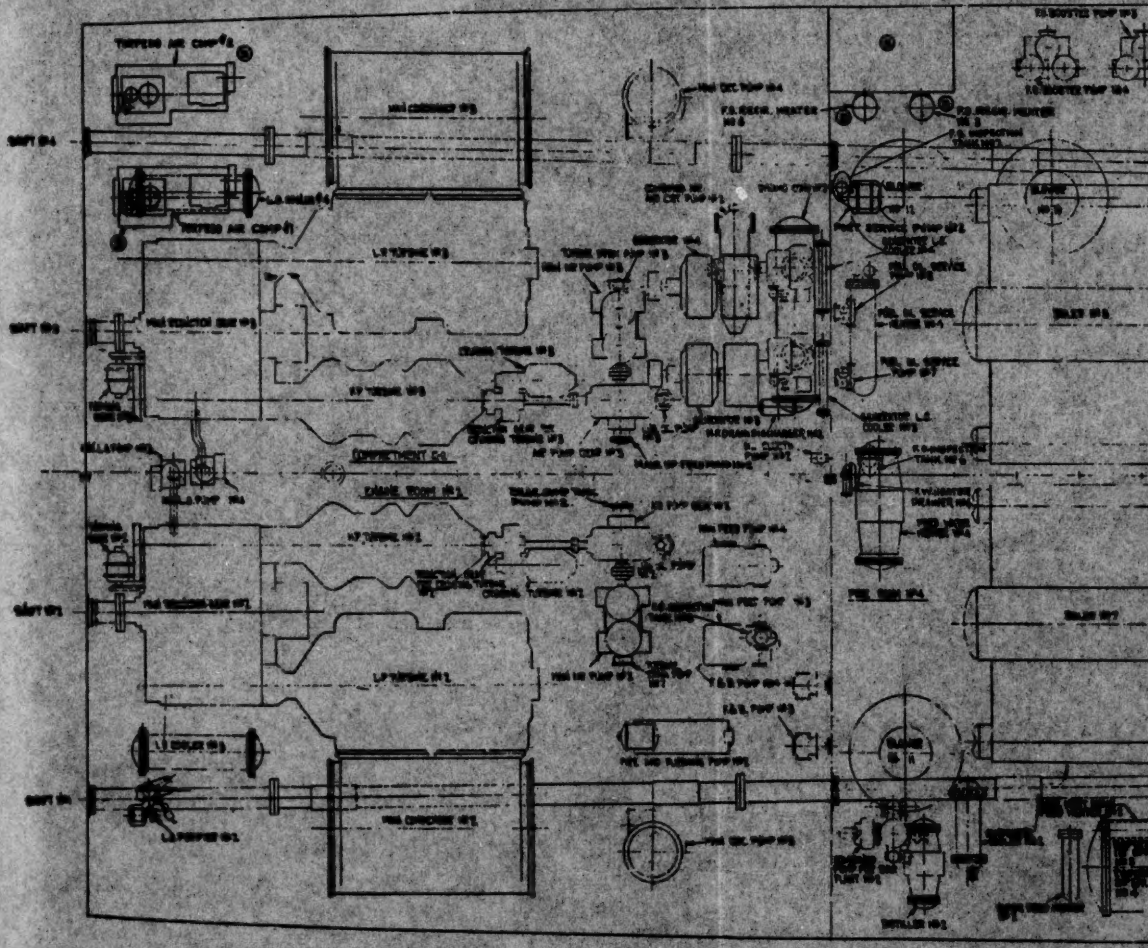
going astern at 200 R.P.M. various steering tests were conducted which were entirely satisfactory.

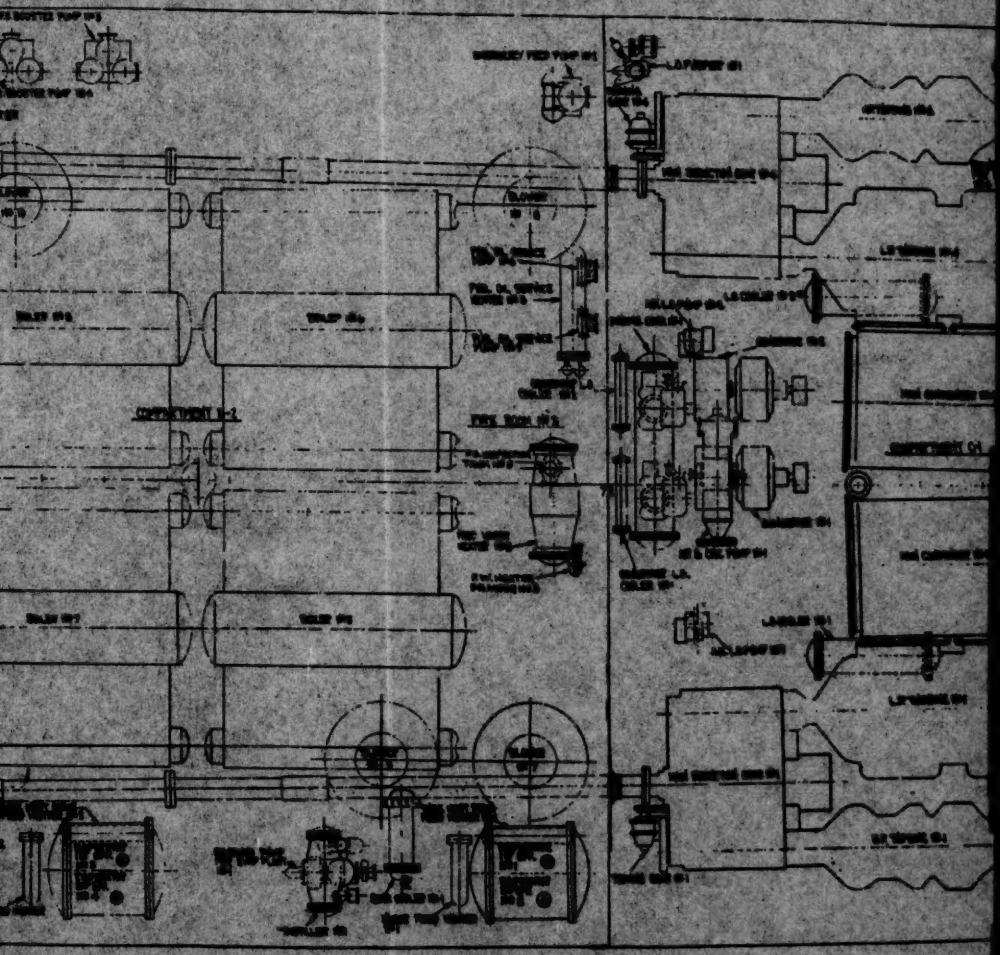
With the ship going astern at 200 R.P.M. the engines were reversed to approximately 300 R.P.M. ahead. The elapsed time after the signal until the ship was dead in the water was 2 minutes 30 seconds. The stern reach was approximately 500 yards. Operating personnel in engine rooms and firerooms conducted these tests very commendably.

The trials completed, the ship proceeded to the builders' works at Camden, N. J.

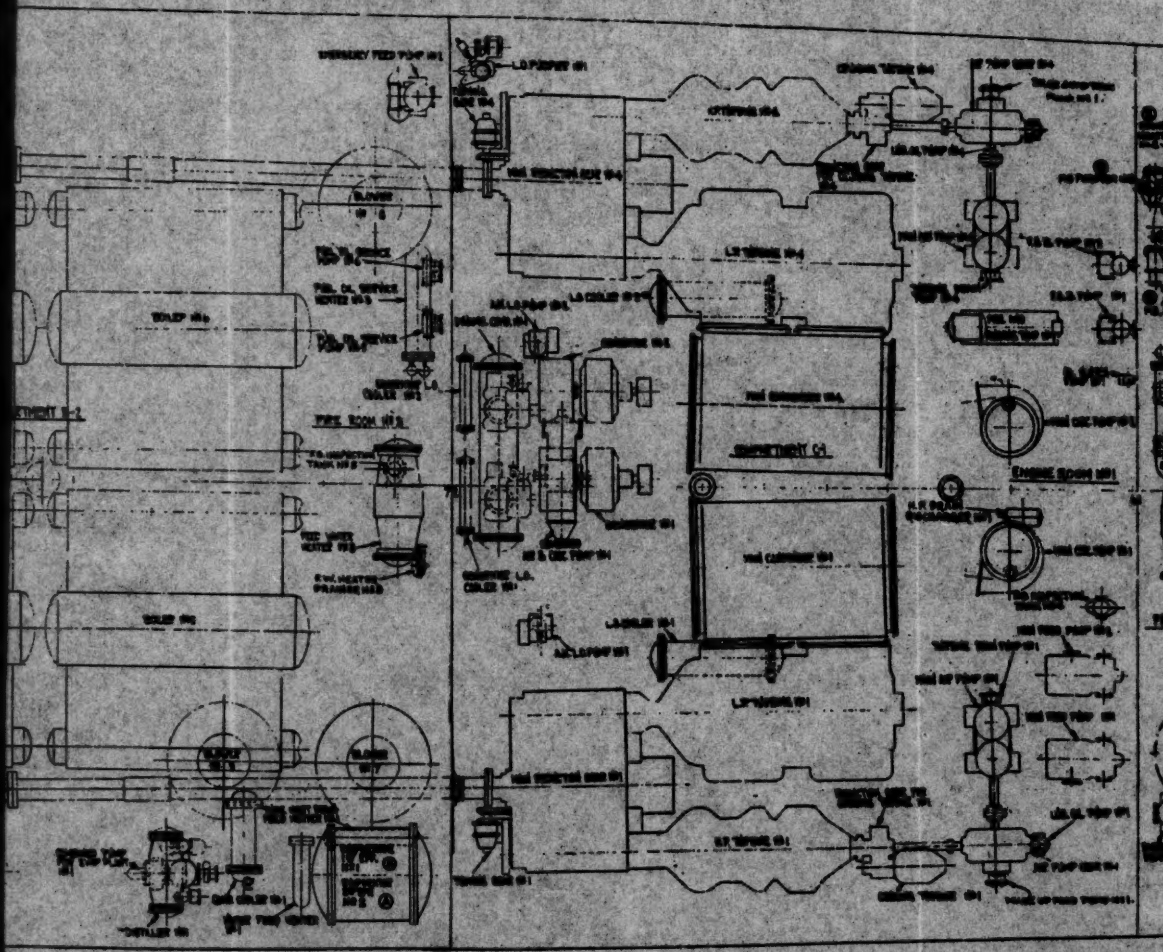
The general comment of the Trial Board was in part as follows:

"The *Salt Lake City* is a fine and able cruiser. The Board has been most favorably impressed by her performances. She steams well and easily; cuts the water like a knife; makes but moderate wave effect; and steers with ease. The ship's motion, in a moderate sea, with sea ahead, astern, quartering and abeam, is easy, and she gives every evidence of being a good and dry gun platform."

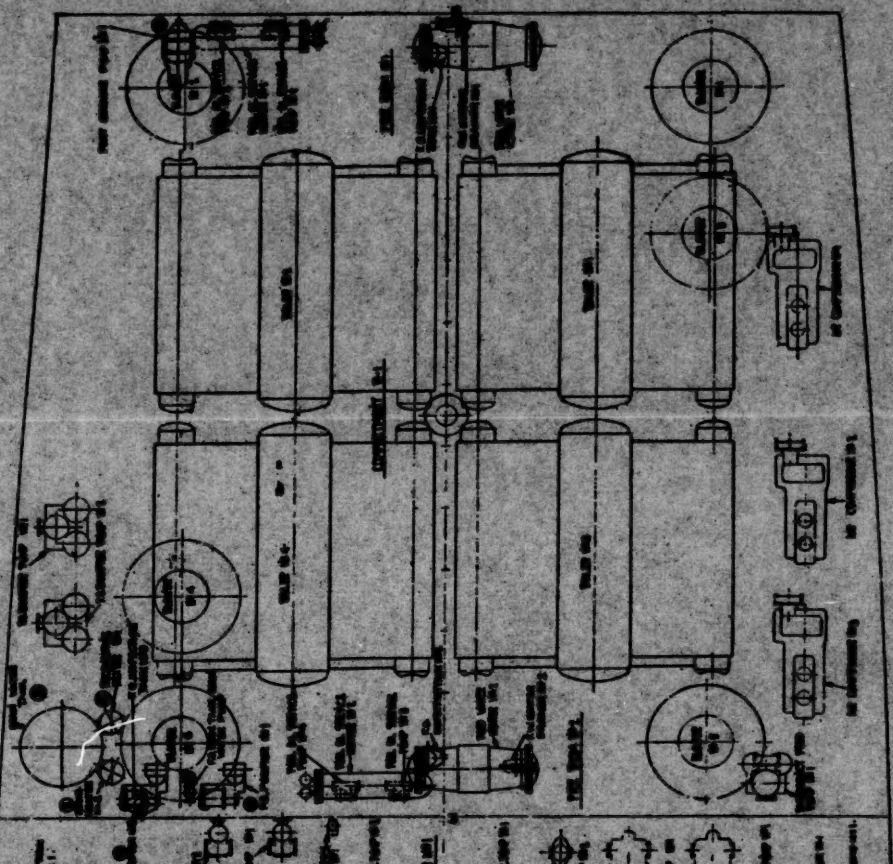


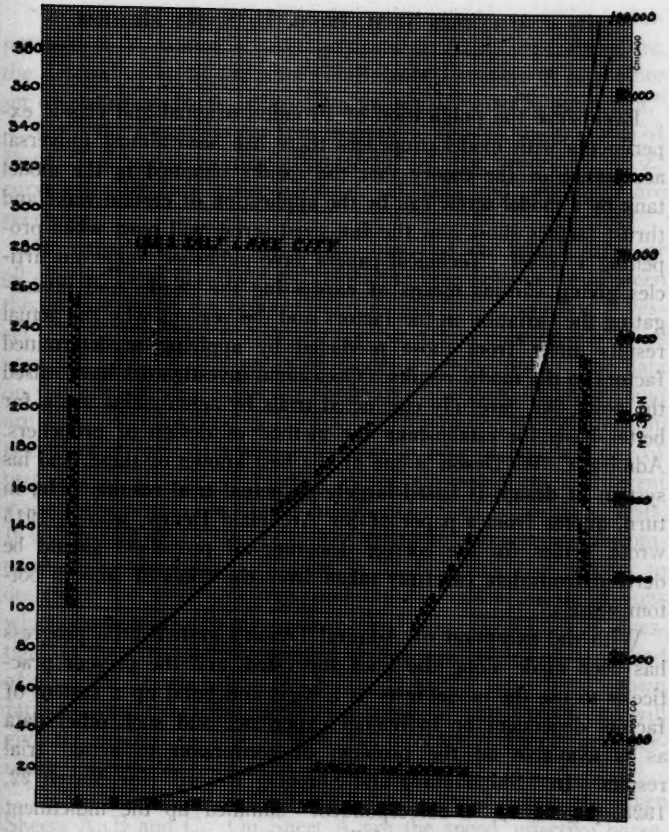


U. S. S. "Salt Lake City"—Arrangement of Machinery.



U. S. S. "Salt Lake City"—Arrangement of Machinery.

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U.S.S. SALT LAKE CITY
CHICAGO
NEW YORK

SIMPLIFIED PROPELLER DESIGN.

By COMMANDER J. M. IRISH, U. S. N., MEMBER.

Ever since the publication of Froude's original and classic experiments with model propellers there has been almost universal acceptance of the theory that the results obtained in the model tank by a model screw can by the application of certain wake and thrust factors show how the full sized screw will react when propelling a vessel. Technical papers have published numerous articles giving various means of expressing the results and investigating the variation in the factors and the reasons why the actual results differ from those predicted by applying predetermined factors to the model results. Occasional investigators have raised the point that carefully conducted trials by actual ships are a far better basis for establishing the method of design of propellers. Admiral C. W. Dyson is one of the best known of these, and his system of design is based largely on actual trial results. He, in turn, quotes from a paper of Sir Archibald Denny, who in 1915 wrote "rules for the correct designing of propellers should be derived from data carefully taken from the trials of smooth bottom vessels."

With the exception of Admiral Dyson's system little progress has been made along the lines suggested, and the present practice is to use the model tests as a basis and build up a system of factors dependent on individual experience and such other data as is available on the problem of conversion to actual trial results. In "Shipbuilding and Shipping Record" for March 22, 1928, page 337, "a correspondent" summed up the indictment against existing systems as follows:

"The accuracy of the dimensions obtained thereby (Froude's 1908 curves) depends on the proper estimate of the wake. It has been clearly shown by the measurements of thrust which have been taken on board ship, that this far exceeded in most cases that

which would be expected from considerations of the estimated resistance of the ship. A great amount of judgment has to be exercised in estimating the H for particular cases, and the information is not always to hand on which to base this estimate.

"In a similar manner the 'wake' value which comes into computation of V, cannot be taken accurately. At the best, therefore, the determination of the dimensions from Froude's or any other set of propeller charts, is largely a matter of personal judgment."

Nor is the Dyson system free from these defects, the selection of Basic Slip, power augmentation factor K, the proper Z_s or Z_p curve are all points where personal judgment and experience are necessary.

An exhaustive study of results obtained on actual ships of all classes shows that there exists definite interdependence between the dimensions of the propeller and its action in propelling a vessel. If laid out on a comparative basis this relationship is as apparent as those of a series of model propeller tests. The data on thirteen different ships are shown on Sheets A, B and C, and are discussed more in detail below. Furthermore, the relationship is evident in spite of a wide difference in type of hull, number of propellers and type of blade. While expressed in a different manner the results are similar to those obtained from model tests. It is apparent that if the results of trials fit into a series of comparative curves, then it is possible by reversing the process to designate the results desired, and thereby select the dimensions which are necessary to give results. Following these lines of investigation there has been developed and is now presented a method of propeller design which, proceeding in logical steps from known facts arrives at dimensions that are based on actual trials and will give the desired results. The actual process is simple and personal judgment is reduced to a minimum.

To demonstrate that such relationship exists there are given Sheets A, B and C. On Sheet A are the speed, effective horsepower per shaft curves for various ships, plotted on logarithmic paper. Sheet B gives speed—R. P. M. curves for the same ships. Sheet C gives R.P.M.—S.H.P. (per shaft) for the same ships.

The E.H.P. data is taken from model tank tests and includes all appendage resistance; the other data is from standardization

trials of the actual ships run under conditions as nearly uniform as possible. While full details of the ship's hull, propellers and performance on trial are given in Table III at the end of the article, in order to emphasize the variety of the types used on Sheets A, B and C, a summary is given here in table I.

TABLE I.

Ship No.	*Type*	*No. of Shafts*	*Actual D + P*	*Max. Speed*	*R.P.M.*	*R.P.M./ Kt*	*Total E.H.P.*
1	Battleship	2	38.64	21.04	124.4	5.91	17799
2	Battleship	4	27.89	21.0	170.5	8.12	20610
3	Merchant	2	30.39	14.95	116.2	7.78	4408
4	Battleship	2	35.54	19.3	123.4	6.4	11600
5	Battleship	4	17.78	21.3	331.5	15.5	18450
6	Scout	4	23.33	34.21	367	10.75	51800
7	Destroyer	2	18.92	35.0	456	13.0	15800
8	Destroyer	2	17.84	28.5	356	12.5	4654
9	Gunboat	2	14.00	13.02	231.8	17.7	585
10	Tug	1	25.64	13.5	123.5	9.15	955
11	Merchant	1	31.5	15.45	108.5	6.95	2408
12	Liner	4	32.17	25.0	182.0	7.28	47000
13	Sub-Chaser	1	11.33	17.77	473.2	26.6	1105

It is not claimed that there is anything unique in the results for the individual ships. It is well known that the effective horsepower varies approximately as the cube of the speed, that the S.H.P. varies approximately as the cube of the revolutions, and that the revolutions vary directly with the speed. But, what is remarkable—is that having plotted the vessels on Sheet A, by consideration of speed and the effective horsepower per shaft, we find, corresponding closely to the relative displacement of these curves, an almost uniform increase in the values of Diameter and Pitch, together with a corresponding decrease in revolutions per knot; and furthermore, that a similar variation is found when the same ships' data are plotted on Sheet B, speed vs. revolutions per minute, and on Sheet C for S.H.P. per shaft vs. revolutions per minute. It leads to the conclusion that, just as a family of curves can be drawn from tests of model propellers representing the laws of model propeller action, in a similar way there can be drawn a series of curves from the data of actual ships trials which would incorporate the laws of full-sized propellers when actually driving

a vessel. These laws would therefore combine the true propeller laws with the effect of the hull and, therefore, eliminate the unknown and variable wake and thrust deduction factors.

After demonstrating that there was a remarkable uniformity in the propeller action in spite of variations in hull, number of propellers, and in blade form and number, it was decided to draw up the design curves from considerations of a standard type and then to establish factors for arriving at variations from the standard. The following was selected as a standard:

Twin screw ships of normal hull lines, with three bladed propellers, having a blade form similar, or nearly similar, to the Dyson standard. The vessels used to establish the design curves had an effective horsepower curve established by the model tank and were those on which progressive trials were conducted over a measured mile. On these considerations the design Sheets I, II, III, etc., were established. For single screw vessels and those with multiple screws, and for propellers with other than standard form, factors have been established for correcting the standard results to those actually obtained. The method of design for twin screw vessels is first described in detail and the methods for other types is shown afterwards.

As the speed—E.H.P. curve is the basis on which the design must be built, further consideration of Sheet "A" must be given. Drawn out to large scale these curves showed that the best average expression was e.h.p. $= v^{3.1} \times$ constant. This expression applies to practically all ships over a wide range; but there is a limiting speed above which the hull resistance changes at a greater rate than $v^{3.1}$. The constant, hereafter designated by "C," together with the designated revolutions per minute, fix and control the sum of the diameter and pitch; similarly with a given value of "C" and the diameter plus pitch, the value of r.p.m. per knot is fixed.

DETERMINATION OF D. & P.

Based on these considerations Design Sheet I was drawn. The ordinates are the constant "C" = effective horsepower $\div$ $v^{3.1}$. The abscissa are D + P = actual diameter + face pitch expressed in feet and the diagonal lines are the values for con-

stant r.p.m. ÷ actual ship's speed in knots. For convenience Sheet I-a gives values of $v^{3.1}$ for values of v. To use Sheet I, it is necessary to know the design speed, corresponding e.h.p. and the desired revolutions per minute If the r.p.m. are not fixed it is only necessary to assume one or more values in order to investigate a range of r.p.m.

RULES FOR USE OF SHEET I.

With speed and e.h.p., calculate constant "C."

With speed and r.p.m., find r.p.m. per knot.

With these two values enter Sheet I and find corresponding values of D + P on base line.

EXAMPLE.

As an illustration of the steps in design, let us take a ship of the dimensions given for ship No. I, table III. It is desired that the vessel make 21 knots at 125 r.p.m. For this speed the model tank curve gives the total effective horsepower with appendages of 17,580.

$$\text{e.h.p. per shaft} = 8790$$

$$v^{3.1} = (21.0)^{3.1} = 12740$$

$$\text{"C"} = \frac{\text{e.h.p.}}{v^{3.1}} = \frac{8790}{12740} = .69$$

$$\text{r.p.m. per knot } \frac{125}{21.0} = 5.95$$

From Sheet I for the value of C and r.p.m. per knot is found D + P = 38.65.

DETERMINATION OF S. H. P.

From an examination of Sheet "C" it is found that for a given propeller the S.H.P. may be calculated by the equations

$$\text{S.H.P. (per shaft)} = \text{constant} \times (\text{R.P.M.})^3 \quad . \quad . \quad (1)$$

for convenience, this expression may be rewritten—

$$\text{S.H.P. (per shaft)} = \left(\frac{\text{R.P.M.}}{K}\right)^3 \quad . \quad . \quad (2)$$

It is also apparent that the value of K is dependent on the propeller dimensions and on the load on the propeller. The propeller dimensions may be designated by D + P, and the load by "C," which is the value used on Sheet I and equals

$$\frac{\text{e.h.p. per shaft}}{V^{3.1}}$$

From these considerations, design Sheet II was drawn, giving the values of K based on D + P and the constant "C."

Therefore to calculate the S.H.P. for a given speed, first determine D + P from Sheet I, as described in Paragraph 8, then with the same value of "C," and this value of D + P, enter Sheet II and pick off the corresponding value of "K."

From equations (2) the S.H.P. (per shaft) is found. Continuing same example as in paragraph 8—

R.P.M. = 125

C = .69

D + P = 38.65 from Sheet I

K = 5.3 from Sheet II

S.H.P. per shaft $\left(\frac{125}{5.3}\right)^3$ = 13,115

Total S.H.P. = 26,330

Total I.H.P. = 28,500 (using mechanical efficiency of 92 per cent).

DETERMINATION OF DIAMETER AND PITCH.

In the transactions of the Institute of Naval Architects, 1926, Prof. C. M. Carter deduced certain formulae based on Froude's 1908 paper. Amongst them is an expression for diameter as follows:

$$D^4 = \frac{\text{Constant} \times H}{B.\ R.^2\ V.} \quad . \quad . \quad . \quad (3)$$

Which may be simplified to D varies as $\frac{\text{e.h.p.}}{v \times R^2}$. (4)

where v is the actual speed of the vessel.

By plotting a large number of actual ships' trials, an excellent agreement was found with equation 4. On Sheet IV this simple curve is given so that for a given e.h.p. speed and r.p.m., the best diameter may be picked out immediately from the calculated value

$$\text{of the diameter constant} = \frac{\text{e.h.p.}}{v \times R^2} \quad \text{where R} = \text{R.P.M.}$$

For example above:

$v = 21.00$

e.h.p. per shaft $= 8790$

r.p.m. 125

r.p.m.^{2} 15625 . . . . . (3)

$$\text{Diameter constant} = \frac{8790}{21.0} \times 15{,}625 = .0267$$

Diameter $= 19.0$ feet from Sheet IV

And Pitch $= 38.65 - 19.0 = 19.'65$

This diameter should of course be checked in order to be certain that it can be installed on the hull with proper tip clearance. The minimum clearance may be taken as 0.1 the diameter. If this cannot be done, a smaller diameter should be used, keeping the D + P at same value.

DETERMINATION OF PROJECTED AREA.

The projected area is found by use of Sheet III. This chart, which is taken up more fully under "Cavitation," is based on the fact that the required area for a given propeller is dependent on the tip speed and the S.H.P. absorbed by the propeller.

T.S. = Tip Speed $= \pi$.D.R.

where D = diameter in feet

and R = revolutions per minute

$$T = \text{absorbed thrust} = \frac{\text{S.H.P. (per shaft)} \times 326}{v}$$

Where v = speed of ship in knots.

$$T/\square'' = \text{absorbed thrust per square inch projected area} = \frac{T}{Pa}$$

The curve on Sheet III indicates the allowable T/□″ to avoid cavitation, but in order to provide for increased resistance, due to fouling of ship's bottom or increased displacement, it is advisable to use for design purposes, 95 per cent of the T/□″, which is indicated as a limit for a given Tip Speed. For tugs or vessels which frequently operate at higher thrusts, this figure should be reduced to .90 or even .80 in extreme cases, depending on the service.

Continuing the example, D = 19.0

r.p.m. = 125.0

$$T.S. = \pi D.R. = \pi \times 19.0 \times 125.0 = 7460.$$

From Sheet III, allowed thrust limit = 16.5

$$.95 \times 16.5 = 15.7$$

$$T = \frac{S.H.P. \times 326}{v} = \frac{13,115 \times 326}{21.00} = 204,000$$

$$\text{Projected area} = \frac{204,000}{15.7} = 13,000 \text{ square inches}$$

Disc area (for D = 19) = 41,000 square inches

$$\therefore \frac{Pa}{Da} = \frac{13,000}{41,000} = .317$$

$$\therefore \text{use } {}^{Pa}/_{Da} = .32$$

COMPARISON OF DESIGN WITH ACTUAL.

The design being now complete, it is interesting to compare the results with the actual propeller, given in Table III—Ship No. 1.

	Design	*Actual*	
$^{Pa}/_{Da}$	.32	.32	
Dia	19′.0	18.65	
P	19.65	19.99	
D + P	38.65	38.64	
v	21.0	21.04	from standardization trial.
r.p.m.	125.0	124.4	from standardization trial.
S.H.P.	28,500	28,400	from standardization trial.

At this point it may be asked whether the propeller so designed is the *best* propeller attainable. There is a choice within rather wide limits where the diameter and pitch may be varied, provided the sum is maintained equal to the value of the D + P found from Sheet I for the given values of speed, e.h.p. and r.p.m. As the diameter is reduced, it will be seen that the Tip Speed is smaller, and from consideration of values on Sheet III, it is consequently necessary to increase the projected area in order to avoid the danger of cavitation. Propellers of widely different dimensions can therefore be designed. In 1925, Admiral Taylor in a paper before the Institute of Naval Architects, "Wake Propeller Coefficients," said, " No matter how he may ring the changes of diameter, pitch, blade area, and blade shape, the designer, where cavitation is not to be provided against, cannot produce a propeller efficiency beyond a limit fixed by extraneous conditions such as wake fraction, thrust deduction and revolutions."

In " Shipbuilding and Shipping Record" for March 22, 1928, " a correspondent," using Admiral Taylor's results, developed for the first time, to the writer's knowledge, the fact that " over a wide range of pitch ratio chosen, the function D + P (diameter + pitch) is practically constant."

These conclusions are incorporated in the design curves of the present article—the Hull fixes the e.h.p.—speed relationship, then for a given or assumed revolutions—no better propeller can be evolved, no matter how the dimensions are juggled. The effect of a large increase in area, is to decrease the efficiency—but this has been shown by all model experiments to be comparatively slight. It may be stated, therefore, that a propeller whose dimensions are determined by the simplified method is as efficient as any that can be designed.

ESTIMATE FOR OTHER THAN DESIGN SPEED.

To estimate the revolutions and horsepower for a speed different from the design point, it is necessary to consider the characteristics of curves shown on Sheets A, B, and C. On Sheet A, it will be seen that while a large number of vessels have a speed —e.h.p. curve with constant slope, some show a distinct break in the curve and at the higher speeds, the slope is not the same as for

the lower speeds. Ships Nos. 6 and 10 illustrate this phenomenon. In other words, the factor "C" for these vessels is not constant over the entire range. On Sheet B, this effect is also very marked, but on Sheet C, the horsepower—r.p.m., for all ships curves are straight lines, the factor "K" is therefore constant even in this type of vessel. Therefore, if the S.H.P. can be determined for the new speed—the r.p.m. can be calculated from the S.H.P. by using same "K" value as for the design point. Admiral Dyson states on page 105 of his 3rd edition, that "If there is available for use the data of actual S.H.P. for any speed v for which the tow rope power = e.h.p., the value of the S.H.P. necessary for any other speed v_1, requiring a tow rope horsepower of e.h.p.$_1$, where the vessel is in the same condition as to displacement, cleanliness of bottom and with similar wind and sea conditions, may be computed at once, the propeller being completely disregarded; the actual horsepowers, shaft and effective, being assumed as basic." The fundamental equations by means of which this computation is done, are,

$$\frac{\text{S.H.P.}_1}{\text{S.H.P.}_2} = 10^z$$

where $z = 1.0414 \log (\text{e.h.p.}_1 \div \text{e.h.p.}_2)$

On Sheet VI, a curve shows the values of 10^z for varying values of e.h.p.$_1 \div$ e.h.p.$_2$, based on the above relationship. We now have all the necessary factors for rapidly computing the horsepower required at some speed other than the design point.

let v_1 = design speed.

v_2 = desired speed.

e.h.p.$_1$ = e.h.p. for v_1

e.h.p.$_2$ = e.h.p. for v_2

S.H.P.$_1$ = S.H.P. calculated for design speed v_1

S.H.P.$_2$ = S.H.P. for v_2

For ratio $\frac{\text{e.h.p.}_2}{\text{e.h.p.}_1}$ enter Sheet VI, and find value of 10^z

Then S.H.P.$_2$ = S.H.P.$_1 \div 10^z$.

Example: For Ship No. 1, having determined S.H.P. = 26,330 for 21 knots (paragraph 9), it is desired to obtain horsepower and r.p.m. for speed of 15.51 knots.

e.h.p. at 15.51 = 6,472 (total).

$$\frac{\text{e.h.p.}_2}{\text{e.h.p.}_1} = \frac{6,472}{17,580} = .368$$

$10^{\frac{2}{3}} = 2.87$ From Sheet VI.

$$\therefore \text{S.H.P.}_2 = \frac{26,330}{2.87} = 9,170$$

$$\text{I.H.P.}_2 = \frac{9,170}{.92} = 9,960$$

As "K" is constant, we can calculate the r.p.m. for this speed from the calculated S.H.P.

"K" = 5.30

S.H.P. per shaft = 9170 ÷ 2 = 4,560.

and r.p.m. = $K \times \sqrt[3]{\text{S.H.P. (per shaft)}}$.

= $5.30 \sqrt[3]{4,560}$

= 87.9

For this speed, the actual I.H.P. on trial was 9,800.

and the actual r.p.m. = 87.4

Knowing the e.h.p. for any other speed, the S.H.P. and r.p.m. can be estimated in a similar manner.

SHIPS OTHER THAN STANDARD.

As stated, the design Sheets Nos. I and II are based on a standard of twin screw ships, with three bladed propellers. It is therefore evident that to apply the results to ships with a different number of shafts or with propellers of other than three blades, certain corrective factors must be utilized. (The rest of the method as outlined above, is unchanged.) It is possible for any designer to analyze a group of similar vessels and arrive at these factors for that particular type. The results given below are those arrived at by the writer from analysis of such ships as were available.

In twin screw ships, the principle variations from the standard is that four (4) bladed propellers are frequently used. This

change results in reducing the actual r.p.m. from that which would be normal for a three (3) bladed propeller. For design purposes, therefore, when the r.p.m. are designated for use with a four (4) bladed propeller, the r.p.m. should be increased by 5 per cent before using Sheet I for finding value of D + P.

A second variation is for twin screw vessels with three (3) bladed propellers, but with full after bodies. These vessels have N.B.C. greater than .70, or are short ships with small midship coefficient and the result is a similar decrease in r.p.m. The designated r.p.m. should therefore be increased by 5 per cent before entering Sheet I.

These variations also have an effect on the value of "K" and after the value of D + P is determined, the value of "K" from Sheet II should in both cases be multiplied by .975.

SINGLE SCREW SHIPS.

While vessels with one propeller have a wide range of body types, the effect of these hull lines on the flow of water to the propeller may be combined into two (2) main groups:

(I) The after body with full lines, and (II) the after body with fine lines.

Under group I are cargo carriers with Nominal Block Coefficient greater than .70, and short ships with small midship coefficients and moderate N.B.C. causing fullness of the buttocks. Under group II are vessels with normal lines and N.B.C less than .70.

In Table II are given the corrective factors for each group to be applied to actual r.p.m. and to "K" in a similar manner to that described.

TABLE II (SINGLE SCREW SHIPS).

Group	Corrective Factor to be Applied to			
	Actual R.P.M. Per Knot		"K"	
	4 blades	3 blades	4 blades	3 blades
I.	1.10	1.070	.88	.97
II.	1.070	1.04	.90	.945

As an example of propeller design for Single Screw Ships, let us take Ship No. 11 in Table III.

This hull has following characteristics:

L — 400 feet
B — 54.0
H — 21.1
N.B.C. — .574
Coefficient of M. S. — .91

This is evidently one of group II described in paragraph 18, having small coefficient of midship section and moderate N.B.C.

The design calls for following characteristics:

Single Screw
Number of Blades — 4
Speed — 15.5 knots
e.h.p. — 2408
r.p.m. — 109

Maximum allowed propeller diameter 15'.5

Reduction gearing between propeller and torsion meters.

Solution:

$$\text{"C"} = \frac{2408}{(15.5)^{3.1}} = .49$$

Corrective factor Table III = 1.07

$$\text{Desired r.p.m. per knot} = \frac{1.07 \times 109}{15.5} = \frac{116.6}{15.5} = 7.53$$

D + P = 31.35 from Sheet I.

$$\text{Diameter Coefficient} = \frac{2408}{15.5 \times (116.6)^2} = .0114$$

∴ Diameter = 15.5 from Sheet IV.

Pitch = 31.35 — 15.5 = 15'.85

From Sheet II for "C" = .49 and (D + P) = 31.35

"K" = 7.5

For single screw, 4 blades, group II corrective factor
$= .90$ (Table II)

$\therefore$ "K" $= 7.5 \times .90 = 6.74$

$$\text{S.H.P.} = \left(\frac{109}{6.74}\right)^3 = 4200$$

Allow 4 per cent for reduction gear and bearing losses.

$\therefore$ S.H.P. $= 4380$ at torsion meter.

T.S. $= \pi$D.R. $= 5315$

$$\text{"T"} = \frac{\text{S.H.P.} \times 326}{\text{v}} = \frac{4200 \times 326}{15.5} = 88{,}200$$

Limiting T/$\square$" from Sheet III $= 10.1$

use $.95 \times 10.1 = 9.6$

$$\therefore \text{ Projected area} = \frac{88{,}200}{9.6} = 9215 \square''$$

Disc area $= 27000 \square''$

$\therefore$ $\text{Pa}/\text{Da} = .34$

Final Design		*Actual*	
Pa/Da =	.355	Pa/Da =	.327
D =	15.5	D =	15.5
P =	15.85	P =	16.0
v =	15.5	v =	15.45
r.p.m. =	109.	r.p.m. =	108.5
S.H.P. =	4380	S.H.P. =	4341

THREE SHAFT SHIPS.

The center screw of three shaft ships should be treated as for single screw ships, the wing screws as twin screw, and the results averaged.

FOUR SHAFT SHIPS.

So far as stream lines are concerned, the outboard screws of four shaft ships are more favorably located than for twin screw ships, while the inboard shafts are working under less

favorable conditions and are, to a certain extent, effected by the water discharged by the wing screws. In certain electric drive ships, the two screws on the same side are held to the same revolutions, as the motors run at a synchronous speed with a single generator. These conditions cause variations in the corrective factors which are applied to standard results from Sheets I and II. They cause a reduction in the r.p.m. and in the value of "K" of various amounts. These corrective factors are listed below:

TABLE II-A (FOUR SHAFT SHIPS).

Corrective factors to be applied to.

	actual R.P.M. per kt.	k
shaft running free . .	1.025	.925
shaft held uniform . .	1.05	.95

There is one exception to these general rules; when the ship is long with fine after body, as in our high speed scouts and airplane carriers, the inner screws have same flow as for twin screw and no corrections are applied to r.p.m. while the correction for "K" reduces to .975.

The use of these curves is not limited to the methods above described. They may be used for estimating the performance of a suggested propeller with a given hull, or to determine the best speed which can be obtained with a given hull, if an engine of known power and r.p.m. is to be used. As the second problem is of more frequent occurrence, the method to be used is outlined below:

There are given, the hull dimensions, the speed—e.h.p. curve, the S.H.P. and r.p.m. of the engines. Assume three speeds at intervals of one knot, in vicinity of desired speed; with the corresponding e.h.p., determine values in each case for "C." Also, for the assumed speeds, determine r.p.m. per knot, using given fixed revolutions. For these points on Sheet I find D + P in each case, and continuing to Sheet II with values of "C" and D + P, find corresponding values of "K" and the resultant S.H.P. Plot

on cross section paper, speed vs. S.H.P. and the best speed for the given S.H.P. can be picked off, also similarly the best D + P. Having the speed fixed, the best diameter, pitch and projected area are determined as outlined above.

BLADE SHAPE.

As the standard adopted for these curves is the Dyson Standard, it is probable that variations from this type may have some effect on the relationship established. Yet, so far as it has been possible to determine from the trials of ships with such non-standard blades, there is little discernable difference as long as the blade approaches the standard oval. For narrow tip blades, and broad tip blades, the revolutions are determined by the D + P, but as the center of effort of the blades is located inside or outside, the .7 radius for standard blades, the S.H.P. absorbed by these types is less or greater by an amount equal to difference of center of effort. But the data is too meager to be conclusive.

CAVITATION.

For a complete exposition of the author's ideas of cavitation and development of Sheet III, readers are referred to "A New Criterion of Cavitation" published in JOURNAL of the A. S. N. E. for November 1929. For the purposes of this article, it is sufficient to say that cavitation is dependent on the S.H.P. absorbed by the propeller, the projected area and the Tip Speed. Sheet III gives a curve of limiting pressures for the various Tip Speeds. If a propeller is designed so that for a given Tip Speed, the pressure based on S.H.P. is below that indicated by the curve, then cavitation will be avoided. If a design is located above this curve, then the r.p.m. will increase by approximately 3 per cent and the S.H.P. will exceed the normal by 10 per cent.

THRUST LIMIT.

Sheet III indicates a limit to the S.H.P. which governs the projected area, but on vessels of the Destroyer and Cruiser class where very high thrusts occur, it has been found that for any given D + P, there is a thrust limit and irrespective of the amount

of area used, the r.p.m. and S.H.P. are greater than normal. This is not due to cavitation, but is simply a limit incident to the method of developing these curves. Sheet V based on D + P, gives the limit of thrust within which Sheets I and II apply; if greater thrust is used, the design point must be taken as the speed which corresponds to the thrust at the given D + P, and for the higher speeds, the S.H.P. and r.p.m. must be calculated in a manner similar to that described in paragraph 15. As the value of e.h.p.$_2$ ÷ e.h.p.$_1$ will in this case be greater than one, the dotted line on Sheet VI should be used and the values of 10^x will be one tenth the scale shown on the sheet.

As an example of calculating S.H.P. and r.p.m. for higher thrusts, there is taken the performance of the *Lexington*, reported by Admiral Dyson in JOURNAL of A. S. N. E. for August 1929.

The propellers were 3 blades, $\frac{Pa}{Da} = .65$ Diameter 14'.885, Pitch 13'.25. Projected area = 16,300 square inches. D + P = 28.135. From Sheet VII, the thrust limit for this value of D + P is 140,000. This corresponds to a speed of approximately 27.9 knots and an e.h.p. of 46,580.

$$\text{Thrust} = \frac{46{,}580 \times 326}{4 \times 27.9} = 132{,}000$$

$$\text{For this point "C"} = \frac{46{,}580}{4 \times (27.9)^{3.1}} = .381$$

r.p.m. per knot = 8.5 Sheet I.

∴ r.p.m. = 237.5

"K" = 9.0 Sheet II.

"K" corr. .975 × 9.0 = 8.78

$$\text{S.H.P.} = 4 \times \left(\frac{237.5}{8.78}\right)^3 = 79{,}600$$

This is for design point, therefore for maximum speed 34.71 knots and e.h.p. 121,830.

$$\frac{\text{e.h.p.}_2}{\text{e.h.p.}_1} = \frac{121{,}830}{46{,}580} = 2.62$$

$10^x = .37$ Sheet VI

$$\text{S.H.P.} = \frac{79{,}500}{.37} = 215{,}000 \text{ (total)}$$

$$\text{For this point "C"} = \frac{121{,}830}{4 \times (34.71)^{3.1}} = .508$$

"K" = 8.75 Sheet II.

"K" corr. = $.975 \times 8.75 = 8.55$

$$\therefore \text{ r.p.m.} = \sqrt[3]{\frac{215{,}000}{4}} \times 8.55 = 323.$$

Estimate

v	27.9	34.7
r.p.m.	237.5	323.
S.H.P.	79,500	215,000

Actual

r.p.m.	237.2	324
S.H.P.	79,600	215,000
Actual Propulsive Coefficient—	.585	.564

SUMMARY OF STEPS IN ESTABLISHING A DESIGN.

(1) Having given the desired speed and effective horsepower, including appendages, the number of shafts and designated r.p.m. —(It will be noted that if r.p.m. are not fixed, designer may select a range for investigation and thereby fix the r.p.m.).

(2) Calculate "C" $= \frac{\text{e.h.p. (per shaft)}}{v^{3.1}}$.

Values of $v^{3.1}$ are given on Sheet I-a.

(3) From "C" and r.p.m. per knot, find value of D + P from Sheet I.

(4) Find value of diameter constant $= \frac{\text{e.h.p.}}{v \times \text{r.p.m.}^2}$

and from Sheet IV, fix value of diameter.

(5) Determine value of pitch from D + P and diameter.

(6) For value of "C" and D + P find "K" from Sheet II.

(7) Determine S.H.P. (per shaft) $= \left(\frac{\text{R.P.M.}}{\text{K}}\right)^3$

(8) Determine T.S. $= \pi$.D.R. and limiting T/$\square''$ from Sheet III, use 95 per cent for factor of safety.

(9) Determine T $= \frac{\text{S.H.P.} \times 326}{\text{v}}$

(10) Determine Projected area $=$ T $\div$ T/$\square''$ in square inches.

(11) Find Disc area $= \frac{\pi\ \text{D}^2 \times 144}{4}$ square inches.

(12) Inches Projected area ratio Pa $\div$ Da.

CONCLUSIONS.

(1) Carefully conducted trials of full sized ships give results from which propeller curves comparable to those obtained in tank tests from models.

(2) Within certain limits, the sum of diameter and pitch of a propeller, taken in conjunction with the design loading, establishes the revolutions per minute, and the required Shaft Horsepower.

(3) The effect of wake and thrust deduction by this method is incorporated into the relationships established by curves based on the actual trials.

(4) These relationships form a simple and accurate method of estimating the proper dimensions of a propeller.

(5) Corrective factors may be computed from converting the standard results into those applying to any special cases.

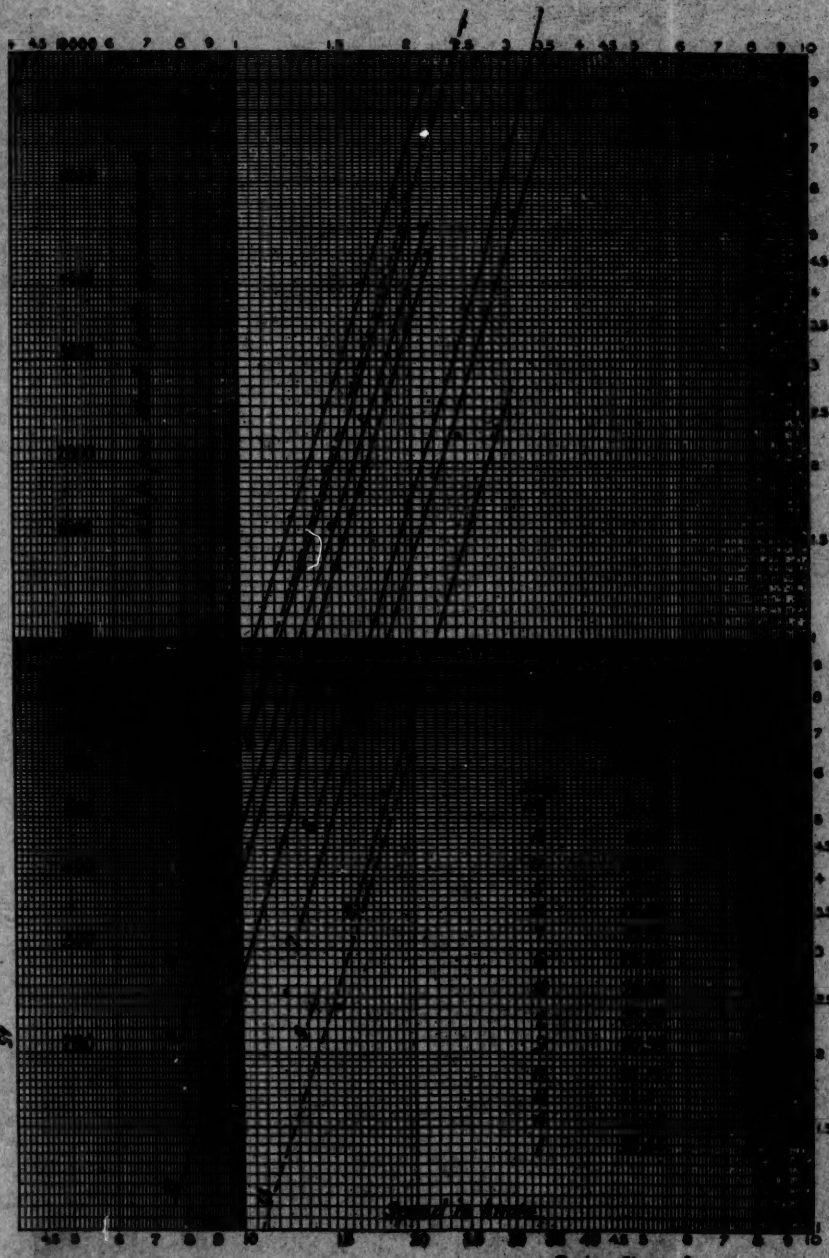
TABLE III.

Ship No.	1	2	3	4	5	6	7	8
LWL	565'	600	528	450	554	550	311	245
B	95.22	96	62.23	80.22	93.2	55.3	30.67	23.5
H	28'.5	30.25	27.5	25.02	27.93	13.8	9.146	6.83
B.B.C.	.61	.649	.726	.633	.622	.604	.469	.41
Coef. of M.S.	.99	.980	.984	.96	.986	.945	.748	.77
B ÷ L	.169	.160	.123	.178	.168	.1005	.0986	.096
No. of Shafts	2	4	2	2	4	4	2	2
No. of Blades	3	3	'3	3	3	3	3	3
Type	Std.	Std.	Std.	Std.	Std.	Std.	Std.	Std.
Pa/Da	.32	.359	.304	315	.501	.605	.594	.358
Dia.	18'.65	13.64	15.95	17.54	9.583	11.33	9.0	7.42
P	19'.99	14.25	14.456	18.0	8.193	12.00	9.917	10.42
Trial v	21.04	21.0	14.95	19.3	21.3	34.21	35.0	28.5
S.h.p. (total)	17,799	20,610	4408	11,600	18,450	51,800	15,800	4654
r.p.m.	124.4	170.5	116.2	125.4	331.5	367	456	356
S.H.P. (total)	-	30,000	7000	17,500	30,000	93,800	27,700	-
I.H.P.	28,400	-	-	-	-	-	-	7400
Problem No.#	36	1	27	35	3	-	40	5

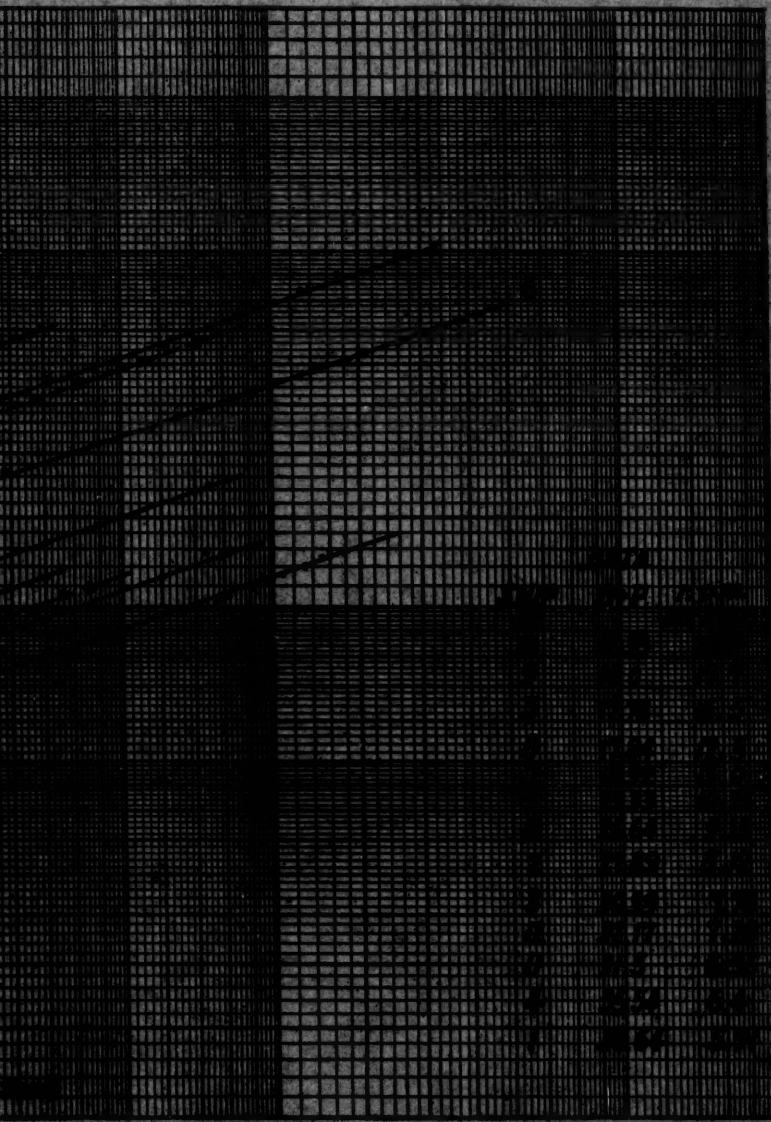
The problem numbers refer to corresponding problems in Adm. Dyson's "Screw Propellers", Third Edition. Where no number is given, ships are from original data.

TABLE III (Continued).

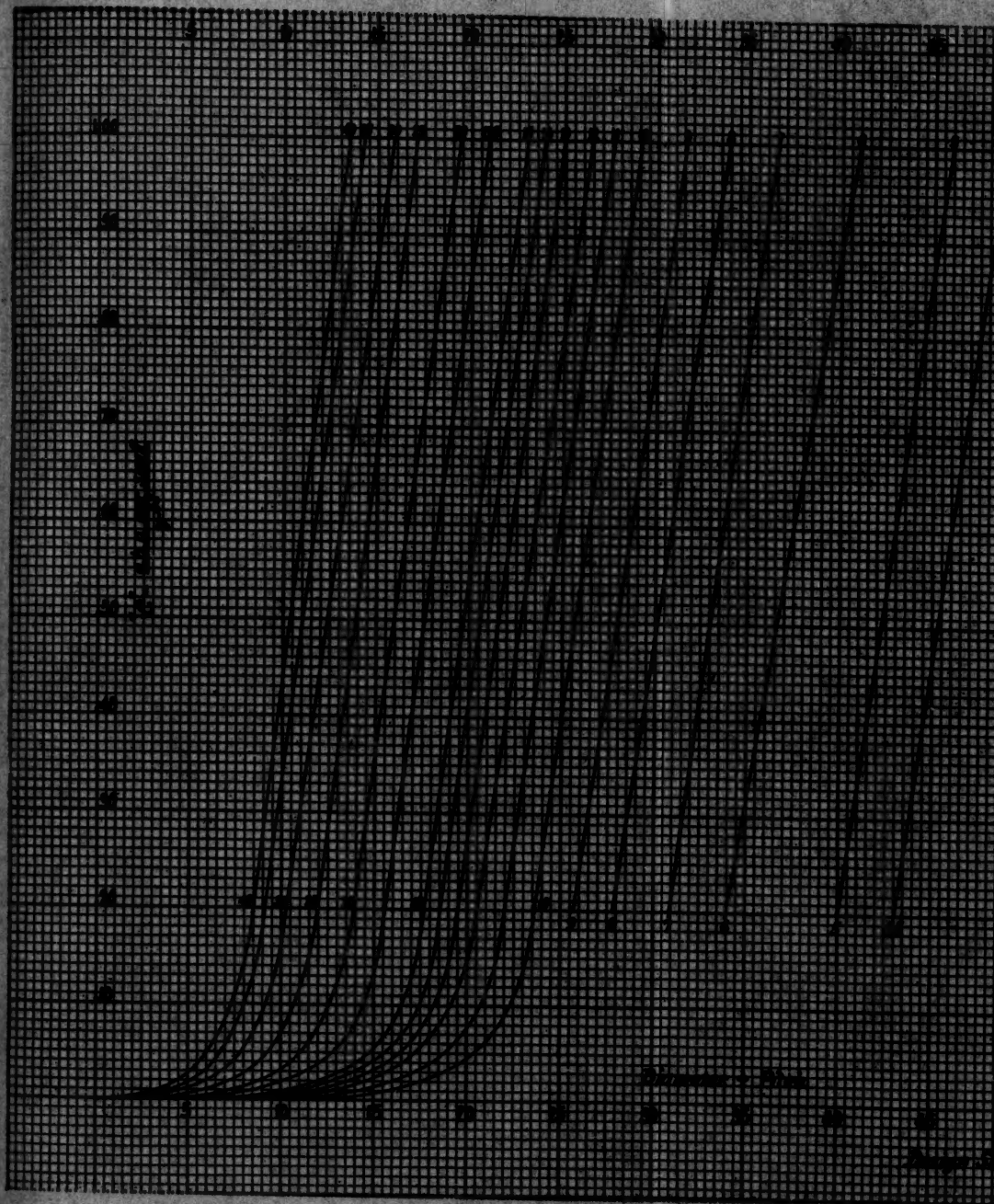
Ship No.	9	10	11	12	13
LLWL	174	185	400	927.5	200
B	34	34.125	54	100	25.5
H	12	12.5	21.1	38	7
M.B.C.	.508	.531	.574	.623	.47
Coef. of M. S.	.858	.875	.91	.97	.80
B ÷ L	.195	.184	.135	.1078	.1275
No. of shafts.	2	1	1	4	1
No. of blades.	3	4	4	3	3
Type	Std.	Std.	Std.	Std.	Std.
Pa/Da	.39	.387	.327	.488	.606
Dia.	6.75	12.86	15.5	16.42	6.58
P	7.25	12.78	16.0	15.75	4.75
Trial v	13.02	13.5	15.45	25	17.77
P_D (total)	585	.955	2408	47,000	1105
r.p.m.	231.8	123.5	108.5	182	473
S.H.P. (total)	-	-	4341	77,300	2079
I.H.P.	1024	1955	-	-	-
Problem No.	17	19	38	-	44

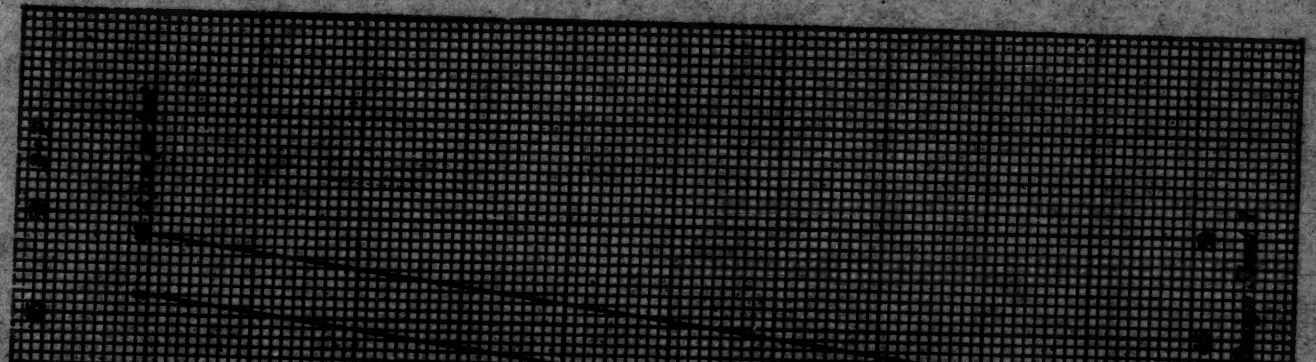


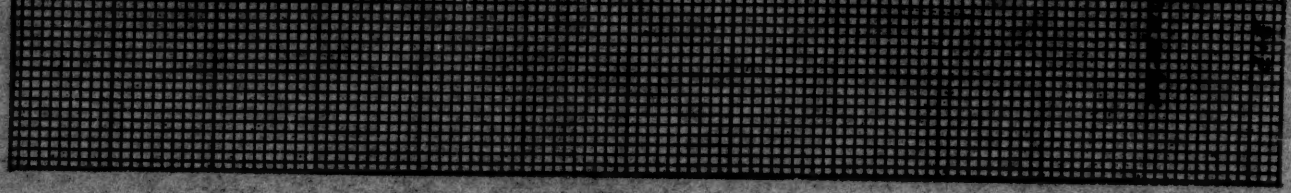
Design Sheet A.

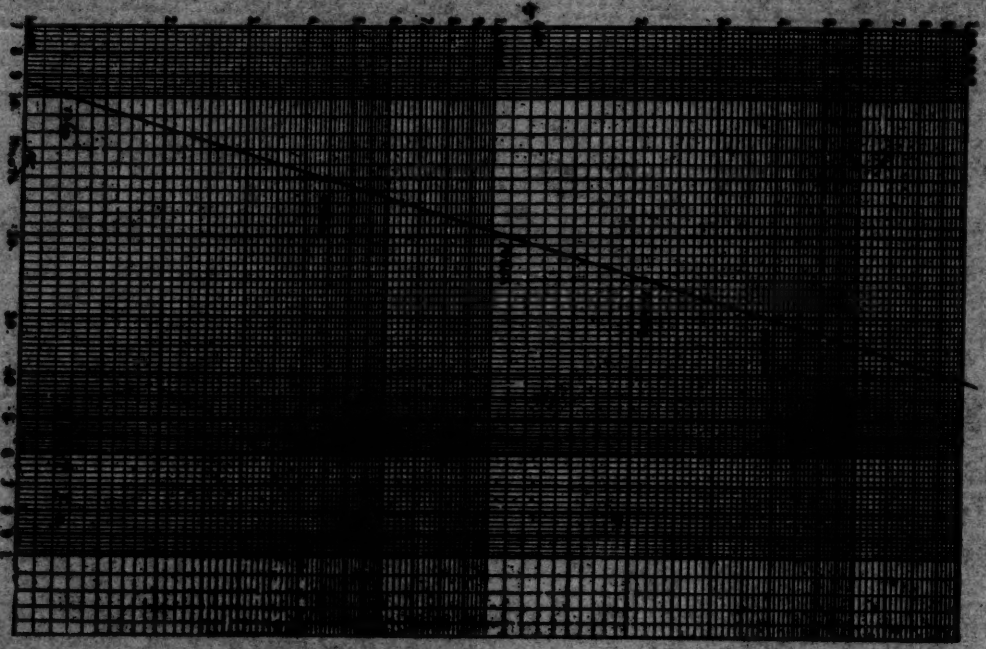


Sheet C.









DIAMETER ÷ PITCH

.50
.40
.30
.20
50
55
60
SHEET II

Scale 75.

Plate III.

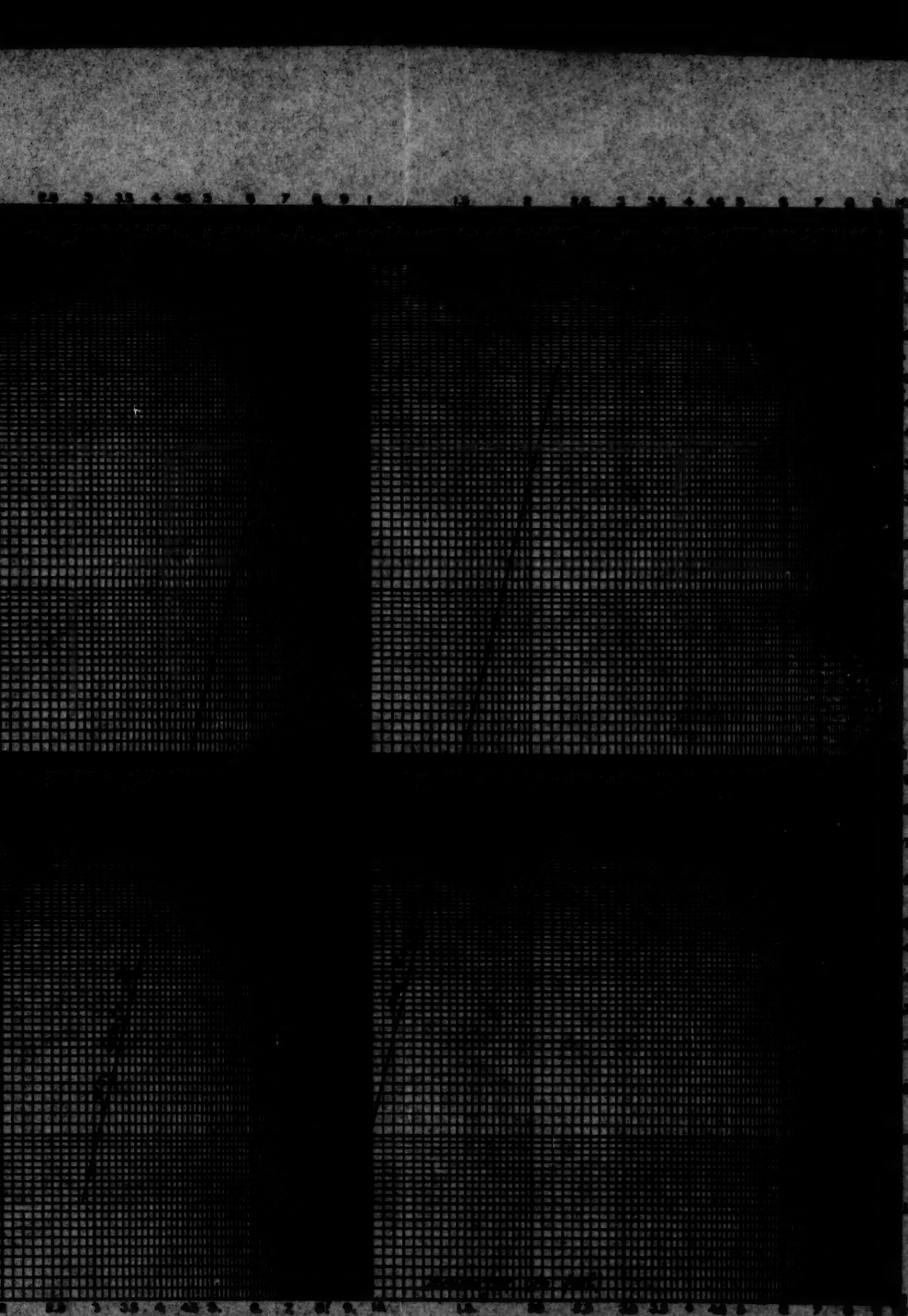
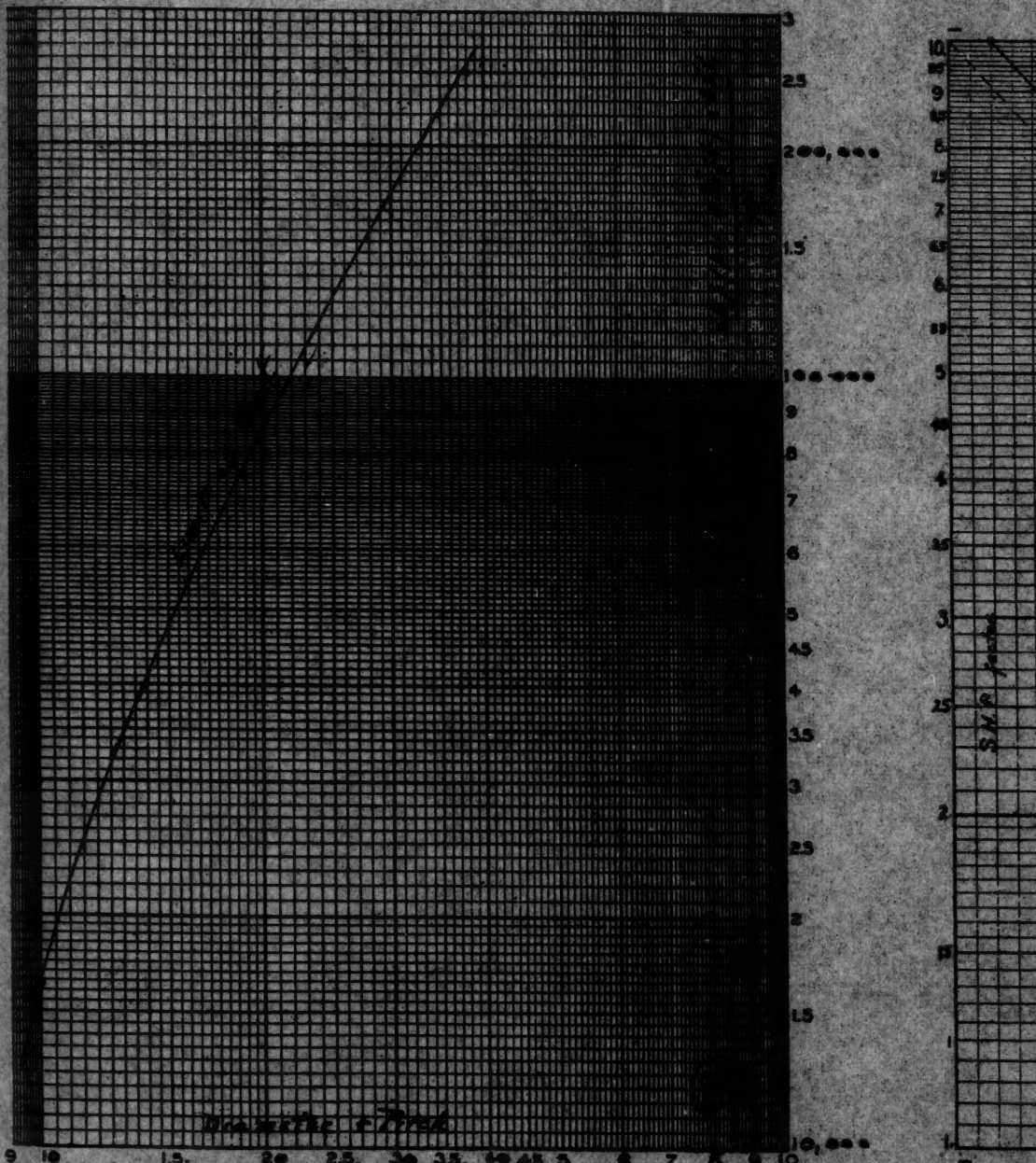
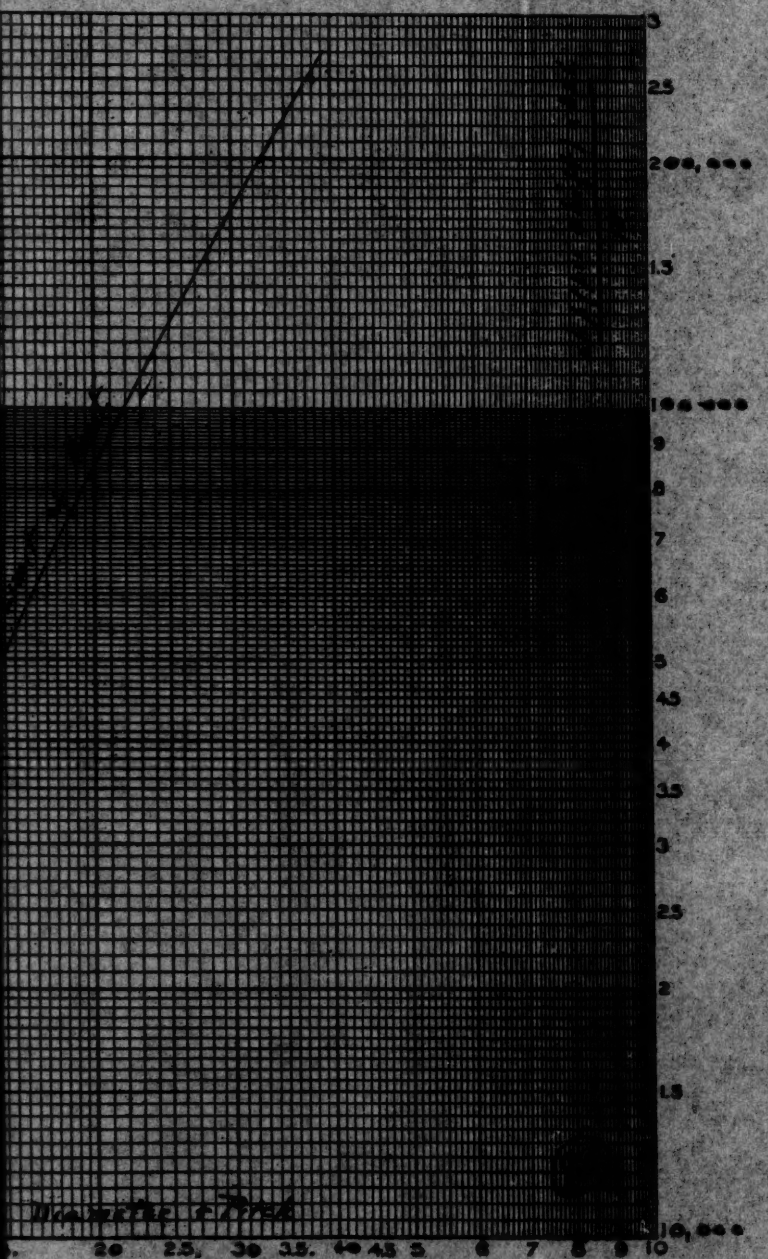


Plate II


Diameter of Pipe
9 10 1.5. 2.0 2.5. 3.0 3.5. 4.0 4.5 5 6 7 8 9 10
3
2.5
200,000
1.5
100,000
9
8
7
6
5
4.5
4
3.5
3
2.5
2
1.5
10,000
10
9
8
7
6
5
4
3
2
1
1

Sheet V.

Sheet II
Speed in RPS


3
2.5
200,000
1.5
100,000
9
8
7
6
5
4.5
4
3.5
3
2.5
2
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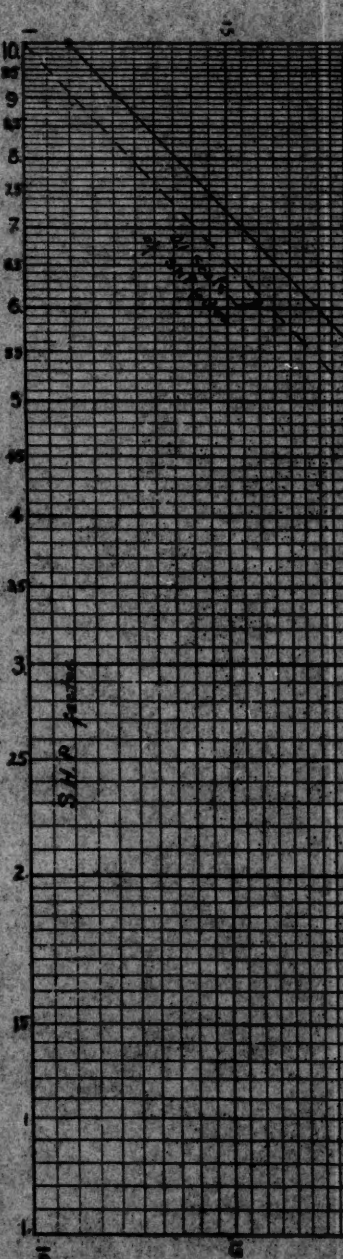
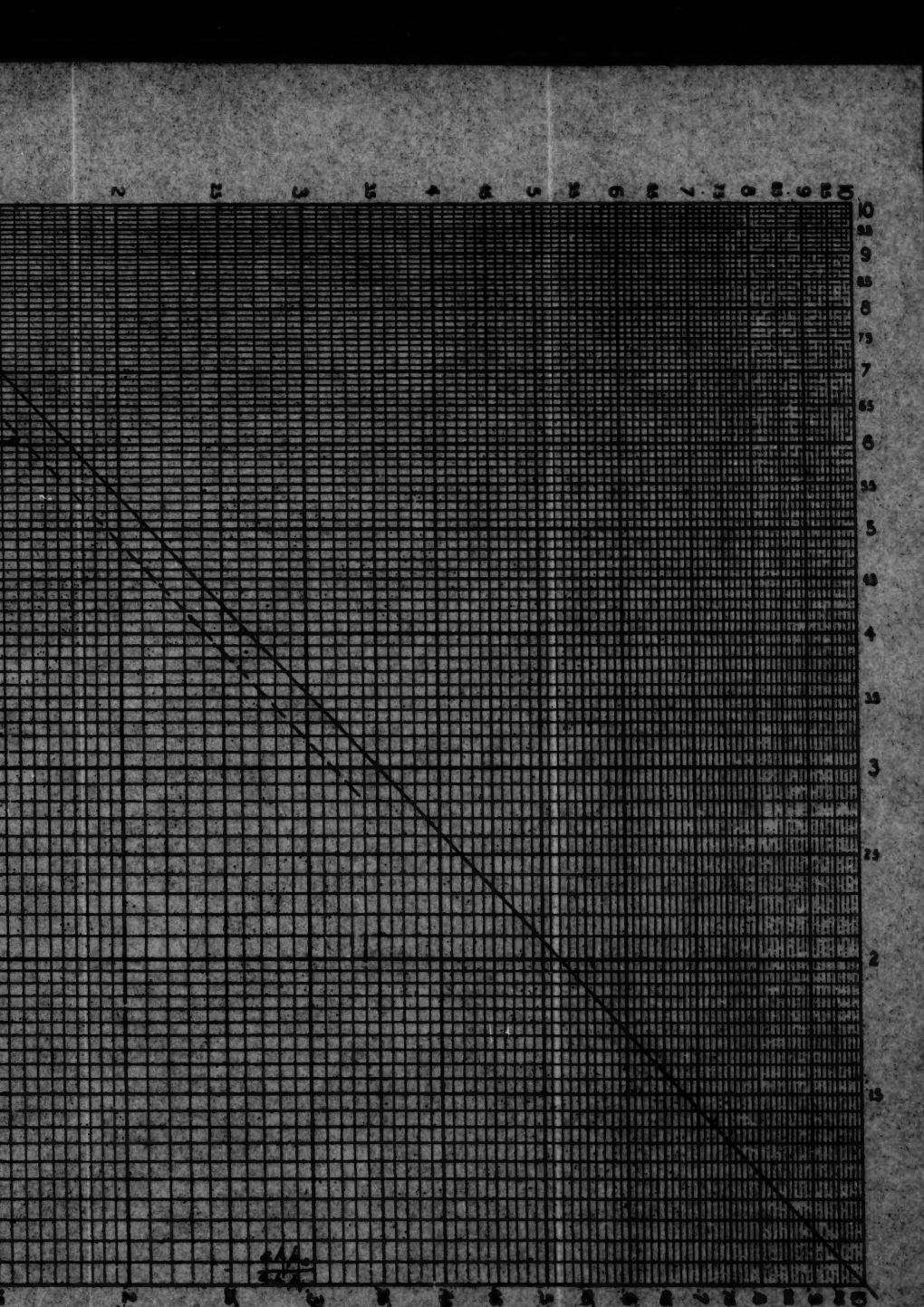
6
7
8
9
10


10
9
8
7
6
5
4
3
2
1

Sheet V.



Sheet II

THE RELATION OF BLOWDOWN TO LIFT AND CAPACITY IN A BOILER SAFETY VALVE.

By Lieut. R. L. Hicks, U. S. Navy.

The setting of the blowdown adjusting ring in a boiler safety valve so greatly affects the lift and capacity that may be obtained from the valve, that it appears desirable to invite attention to this particular. The importance of this adjustment is great, particularly because it is largely under the control of the operating personnel, who may or may not have detailed information concerning the functioning of safety valves.

For the purpose of discussion one type of valve will be used. It should not be understood, however, that any remarks made apply to this type of valve alone.

The valve shown in Figure I is of a well known type. It may be described as a vertical, spring loaded, conical seated, boiler safety valve. The popping pressure is governed by a large spring above the valve. The blowdown is governed by the setting of the ring about the valve seat. The ring is moved up to increase blowdown and vice versa.

The capacity of the valve, under given conditions, is dependent on the valve lift. In testing valves the lift is usually measured and the capacity computed from the lift obtained.

At the Fuel Oil Testing Plant in the Philadelphia Navy Yard, the lift is measured by means of the apparatus shown in Figure II. The popping lift is obtained by the micrometer, Mark A, which measures the position of the pin, Mark C, before and after popping, the difference being the lift. The pin is actuated directly by the valve stem. The lift gauge, Mark D, is used to obtain steady lift data, the lift being read over a series of ascending and descending pressures while the valve is blowing.

Figure III is a series of pressure-lift curves, made while the valve was blowing, using a valve of the type shown in Figure I, and the apparatus shown in Figure II. For this particular series

of curves, popping pressure was set at 310 pounds gauge, and the blowdown adjusting ring varied, the normal specified blowdown being 9.3 pounds. These sustained lift curves have approximately the same slope, an increase in blowdown resulting in an increase in valve lift over the whole range of operating pressures. These representative sustained lift curves also serve to illustrate how greatly the lift of the valve is reduced when the blowdown is set to less than that designed.

The steam discharge opening corresponding to any lift of the valve in a valve of the type shown in Figure I (with a conical seat of approximately 45 degrees) may be computed from the formula.

$$\text{Area} = 1.111 \times a\ [2b + a]$$

Where a = Vertical lift of valve.

b = mean contact diameter of disc.

This formula indicates that the steam discharge area increases with an increase in blowdown, at a rate nearly proportional to the increase in lift.

The capacity of the valve under any condition within the range covered by the curves of Figure III may be determined by computing the discharge area from the measured lift, and the discharge capacity corresponding to that area from the following modification of Grashof's formula.

$$\text{Lb./hr.} = \text{Area} \times \frac{3600}{61} \times p^{.97} \times c$$

Where p = absolute pressure at base of valve.

c = discharge coefficient for the particular series of valves.

In the absence of better data, a coefficient of .95 may be assumed for a valve of the general character of Figure I.

That is, the capacity of the valve when blowing at any steam pressure, is nearly proportional to the lift, which is dependent on the blowdown setting.

It is the present practice on oil burning water tube Naval boilers to install safety valves of such an emergency capacity that, when set to 3 per cent blowdown, they will relieve the total capacity of the boiler at not over 25 per cent above the popping pressure.

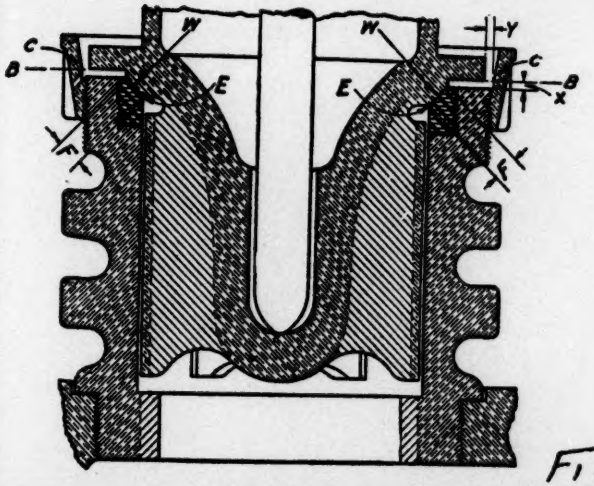
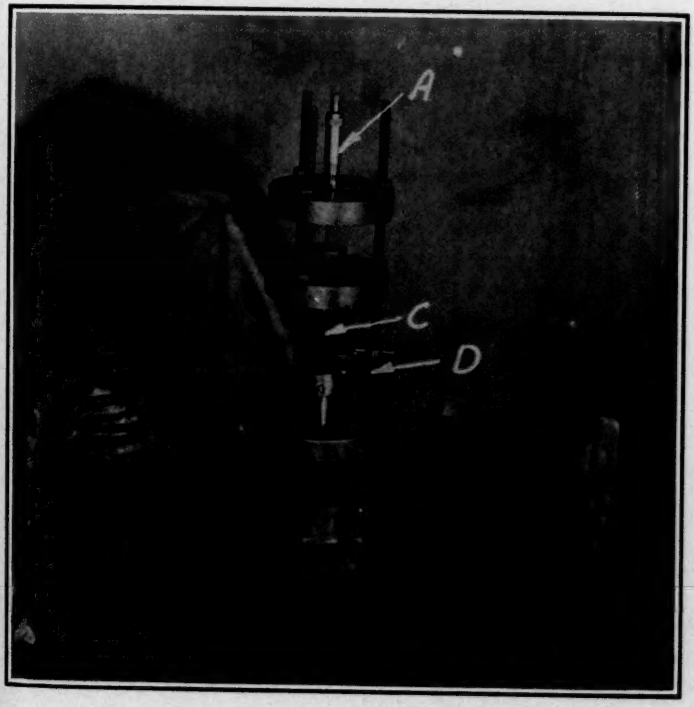


Fig I.



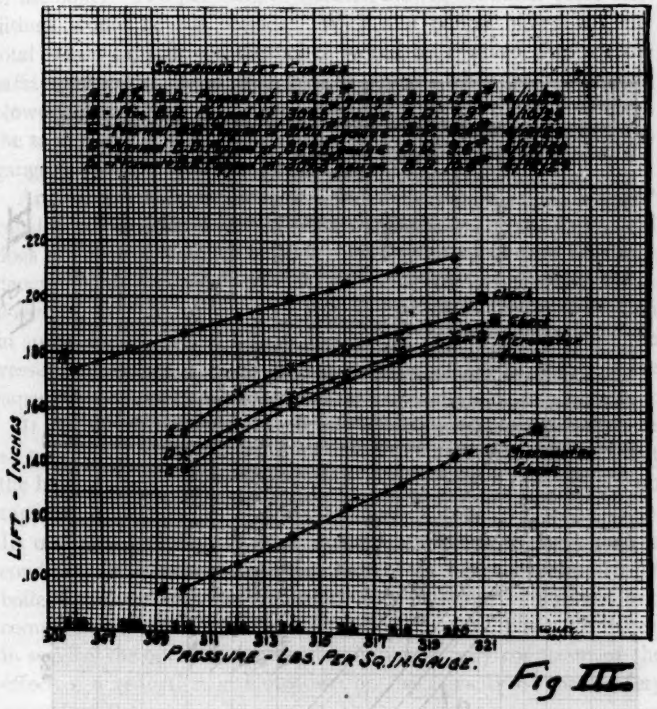


Fig III.

It should be noted especially that the valves will not relieve the total steam capacity of the boiler at popping pressure, and, at a blowdown of less than 3 per cent, may not relieve the total steam capacity of the boiler until the boiler pressure is above the usual hydrostatic test pressure.

To illustrate the boiler safety valve capacity conditions which may obtain on a boiler with the present Naval installation practice, a characteristic series of curves have been computed and are shown in Figure IV. This series of curves assumes a working pressure of 310 pounds gauge, a valve of the general type shown in Figure I, and the combustion rate of 1 pound of oil per square foot boiler water heating surface as the maximum steaming rate

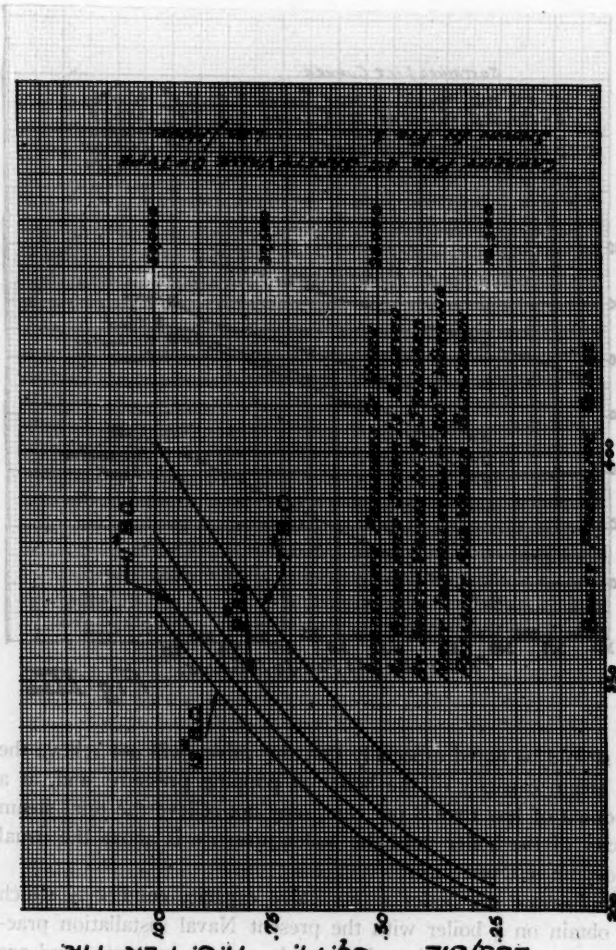


Fig XI

of the boiler. A 3 per cent, or normal, blowdown under these conditions would be 9.3 pounds. With a 9 pound blowdown, the total steam capacity of the boiler would be relieved through the safety valves at about 380 pounds gauge, while at a 7 pound blowdown the total steam capacity would not be relieved through the safety valves until boiler pressure had reached over 400 pounds gauge.

Among operating personnel, a boiler safety valve is generally considered satisfactory if it pops regularly, seats regularly, and does not chatter, simmer or leak. There also appears to be a tendency among fireroom personnel to reduce the safety valve blowdown, in order to save fresh water or fuel, or to avoid a reduction in speed due to lowered steam pressure, if the valves lift while the vessel is steaming at high speeds. The question of safety valve capacity is seldom considered.

It is conceded that the firing of boilers will probably never be so poor, or conditions outside the fireroom will never be such that the boiler safety valves will be required to relieve the total steam capacity of the boiler. However, such a condition is possible. In order to maintain a valve capacity sufficient to meet such a condition, it appears desirable that, after overhaul, the lift of boiler safety valves be measured under normal conditions, and compared with the lift shown on the valve name plate. Further, in service, the operating personnel should be fully cognizant of the effect of a reduction in blowdown on the capacity of their safety valve installation.

SOME CONVENIENT HEAT TRANSFER FORMULAS APPLYING TO FUEL OIL HEATERS.

By Lieut. G. M. Dusinberre, U. S. Navy, Member.

If a fuel oil heater becomes dirty, it may still perform satisfactorily at low ratings, but under full power conditions may fail to raise the temperature of the oil to the required point. In the coil type heater the oil side of the heating surface cannot be inspected, and in any type, breaking down for inspection and cleaning is a large job and involves the probability of trouble in making up the joints. The object of this paper is to develop formulas by which the condition of the heater can be determined at any time from a set of readings of the available instruments, and by which the performance of the heater can be predicted for full power conditions. These formulas have been checked on tubular heaters of the coil and straight tube types, but may not apply to film type heaters.

The formulas will first be stated in their working form, with examples. Then the derivation of the formulas will be given, and their limitations and probable errors will be discussed.

Following is the data to be obtained for a set of readings:

G = Gallons fuel oil per hour, from meter readings, applying meter correction if known.

t_i = Temperature of oil at heater inlet, from thermometer.

t_o = Temperature of oil at heater outlet, from thermometer.

p_g = Pressure of steam in heater shell, from gauge.

t_s = Temperature of steam in heater shell, taken from steam tables for the absolute pressure corresponding to the observed value of p_g.

It is desirable to check all instruments before taking readings. Care should be taken that the heater is properly drained and vented of air.

Take two sets of readings under steady conditions, at different values of (G). Indicate the set of readings for the smaller

value of (G) by the subscript (1), and for the larger value by the subscript (2). (G_2) should preferably be at least twice as large as (G_1). Example:

	(1)	(2)
G	300	600
t_i	88	86
t_o	190	190
p_g	63	129
t_s	310	355

In this and the following examples, observed quantities are taken to the nearest whole number, and calculations are by slide rule.

For each set of readings, solve for (Z) in the equation:

$$Z = \log 10 \left(\frac{t_s - t_i}{t_s - t_o} \right) \quad . \quad . \quad . \quad (1)$$

Example (for first set of readings):

$$Z_1 = \log 10 \left(\frac{310 - 88}{310 - 190} \right)$$

$$= \log 10 \left(\frac{222}{120} \right)$$

$$= \log 10 \, (1.850)$$

$$= .267$$

Similarly, $Z_2 = .212$

Then solve the equation:

$$m = \frac{\log (Z_1/Z_2)}{\log (G_2/G_1)} \quad . \quad . \quad . \quad (2)$$

Example:

$$m = \frac{\log (.267/.210)}{\log (600/300)}$$

$$= \frac{\log 1.260}{\log 2.000}$$

$$= \frac{.100}{.301}$$

$$= .332$$

The value of (m) thus obtained may be regarded as a constant for the heater, regardless of its condition as to cleanness.

In a large number of tests on Reilly multicoil heaters and in a few on Griscom-Russell straight tube heaters, (m) was found to have values between .32 and .36. If it is desired to predict the performance of the heater from a single set of readings, (m) may be taken as 1/3.

From either set of readings, solve for (Y) in the formula:

$$Y = ZG^m \qquad (3)$$

Example (from first set of readings):

$$Y = .267 \times 300^{.332}$$
$$= .267 \times 6.64$$
$$= 1.774$$

The value of (Y) thus obtained is taken as a constant for the heater in its present condition.

We can now predict the performance of the heater under full power conditions. Assume that 2100 gallons of oil per hour will pass through the heater, entering at a temperature of 84 degrees, and that the maximum available steam pressure on the heater shell will be 225 pounds. (On account of the line drop, full boiler pressure may not be obtained at high rates.) Then:

$$G_3 = 2100$$
$$t_{i3} = 84$$
$$p_{g3} = 225$$
$$t_{s3} = 397$$

and t_{o3} is to be calculated.

Substituting in formula (3):

$$1.774 = Z_3 \times 2100^{.332}$$
$$Z_3 = \frac{1.774}{12.67}$$
$$= .1402$$

Substituting in formula (1):

$$.1402 = \log 10 \left(\frac{397 - 84}{397 - t_{03}} \right)$$

and solving:

$$t_{03} = 170$$

It is seen that if 190 degrees is required for satisfactory atomization, then this heater should be cleaned before attempting a full power run.

The derivation of the formulas begins with the definition of the overall heat transfer coefficient as B.T.U. per hour, per square foot heating surface, per degree F. of mean temperature difference, or:

$$K = \frac{\text{B.T.U.}}{(\text{hr.})\ (\text{sq. ft.})\ (\text{M.T.D.})} \quad . \quad . \quad (4)$$

In a fuel oil heater or similar apparatus:

$$\frac{\text{B.T.U.}}{\text{hr.}} = \frac{\text{gal.}}{\text{hr.}} \times \frac{\text{lbs.}}{\text{gals.}} \times (\text{sp. heat}) \times (t_o - t_i) \quad (5)$$

The product of the second and third terms on the right is the heat capacity per unit volume, or B.T.U. required to raise one gallon one degree F. For simplicity in the formulas it is necessary to assume this as a constant, and when figuring on the same oil between the same temperature limits, no error is introduced by such an assumption. When oil is heated the density decreases, but the specific heat increases at a more rapid rate. The heat capacity per gallon of an oil passing through the meter at 80 degrees differs by only 1.6 per cent from the value for the same oil at 60 degrees. For oils differing in specific gravity by 4 per cent, the heat capacity per gallon differs by only 0.9 per cent. But due to the effect of viscosity on the characteristics of the heater, it would probably not be safe to attempt to predict its operation on Bunker "C" oil from data obtained with Bunker "A," or vice versa. With sufficient data, analyzed by the method presented here, some relation might be established between the performances of a heater on oils of different viscosities.

In the type of apparatus under discussion, the mean temperature difference is given by the formula:

$$\text{M. T. D.} = \frac{(t_o - t_i)}{\log e \left(\frac{t_s - t_i}{t_s - t_o}\right)} \quad . \quad . \quad (6)$$

To simplify slide rule calculations we can write:

$$\text{M. T. D.} = \frac{0.434\ (t_o - t_i)}{\log 10 \left(\frac{t_s - t_i}{t_s - t_o}\right)} \quad . \quad . \quad (7)$$

Substituting equations (7) and (5) in equation (4), and letting:

$$C_1 = \frac{\text{lbs.}}{\text{gal.}} \times \frac{\text{(sp. heat)}}{0.434\ \text{(sq. ft. heating surface)}} \quad . \quad . \quad (8)$$

we obtain:

$$K = C_1 \times G \times \log 10 \left(\frac{t_s - t_i}{t_s - t_o}\right) \quad . \quad . \quad (9)$$

The heat transfer coefficient varies in a rather complex way with viscosity, velocity, turbulence, and other factors. There is an empirical formula, used in condenser calculations, which expresses the net result of these effects with sufficient accuracy for engineering purposes. This formula is:

$$K = CV^n \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where V is the mean velocity of the cooling medium and C is a constant.

As an oil heater is in effect a condenser using oil as the cooling fluid, this formula should apply. The mean velocity of oil flow is proportional to (G), so we can write:

$$K = C_2G^n \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

This was found to hold good over the range from cruising to full power conditions on destroyers, for a single grade of oil.

Further development depends on the assumption that (n) is constant for a particular design of heater, and independent of the degree of fouling of the heating surfaces. This was found to be substantially true for the conditions mentioned in preceding paragraph.

On this assumption, C_2 is a function of the condition of the heating surface. We shall define the efficiency of a heater as the

ratio of this constant to the corresponding value for a heater known to be perfectly clean. That is:

$$E = C_2/C_3 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

where C_3 is the value of C_2 for a new or perfectly clean heater. But, from equation (11):

$$C_2 = K/G^n$$

and for a given set of conditions, by equation (8):

$$K = C_1 \times G \times \log 10 \left(\frac{t_s - t_i}{t_s - t_o}\right)$$

Substituting in equation (12):

$$E = \frac{C_1 \times G \times \log 10 \left(\frac{t_s - t_i}{t_s - t_o}\right)}{C_3 G^n}$$

Letting $C_4 = C_1/C_3$ and $m = 1 - n$, and simplifying:

$$E = C_4 \times G^m \times \log 10 \left(\frac{t_s - t_i}{t_s - t_o}\right) \quad . \quad (13)$$

By substituting a set of readings for a heater known to be perfectly clean, and setting $(E) = 1$, the value of (C_4) can be calculated. Within limits for which (C_1), (C_2), and (n) are constants, (E) is a constant depending only on the cleanness of the heating surfaces, it is the ratio of the heat transfer coefficient of the heater to that of a clean heater at the same rate of oil flow, and it can be used to predict the performance of the heater. Simply for purposes of prediction it is not necessary to know the value of (C_4). In equation (3), the arbitrary symbol (Y) represents E/C_4, therefore (Y) is also a constant proportional to (E). Equation (2) is easily derived from equation (3).

Possible errors in the formulas have been discussed and it remains to consider the effect of observational errors. If it is desired to obtain actual values of the heat transfer coefficient or of the heater efficiency, the effect of these may be considerable. For purposes of comparison and prediction it is less important, so that even without special precautions one may expect predictions to be accurate within about 5 degrees. A constant percentage error in (G) has no effect on (t_{o3}). The effect of an error in (m) is

almost negligible. In the examples given above, a 1 per cent error in (m) results in about 1/2 degree error in (t_{o3}). A 2 degree error in (t_{s1}) or (t_{i1}) affects (t_{o3}) by only 1 degree, while an error in (t_{o1}) gives about an equal error in (t_{o3}).

Another field of usefulness of this method of analysis is that it gives a definite measure of the effect of different methods of cleaning. For example, a number of coil type heaters on destroyers showed efficiencies ranging from .45 to .60. This was by comparison with the manufacturer's estimated performance for a new heater. The results of various methods of cleaning were as follows:

Method of cleaning	*Resulting Efficiency*
Boiling out with kerosene and compound as prescribed in M. E. I.	.64
Breaking down heater and burning out coils with oxygen	.80
Circulating hot kerosene at intervals for two days, with handy-billy	.78

For straight tube type heaters, it would be desirable to compare the effect of wirebrushing the tubes.

The writer is indebted to Mr. A. P. Weckerly of the Bureau of Engineering for constructive comment in the preparation of this article.

THE INSTALLATION OF PLASTIC FIRE BRICK MATERIAL.

By Lieut. R. L. Hicks, U. S. Navy.

The expansion of the Navy, during and subsequent to the World War, resulted in a greater variety of boiler types and sizes, and a consequent greater number of special refractory shapes necessary to line the furnaces of these boilers. The cost of these special shapes, together with the problem of maintaining proper supplies in the particular localities where they were in demand, led to various investigations to obtain a substitute. As the result of these investigations, largely made at the Fuel Oil Testing Plant at the Philadelphia Navy Yard, in collaboration with various refractory manufacturers, a "plastic fire brick material" was developed, which could be moulded on the job, to the particular shape required. This led to an elimination, in the Navy, of over 1100 special refractory shapes. The manufacture and use of plastic fire brick material has been extended during the last few years, until excellent material of this type may be obtained from a number of manufacturers, and the material is now in common use commercially.

Plastic fire brick is essentially unburned fire brick. The chemical analysis and P.C.E.* of the plastic material is nearly the same as the fire brick which is manufactured from the same raw material. The material is received in a stiff plastic condition, the moisture content being about 11 per cent by weight. A moisture content of less than 10 per cent makes the material somewhat difficult to work, while with a moisture content of over 13 per cent, the material tends to become "muddy."

The material may be used in any place in the furnace lining where a standard 9 inch shape cannot be used. It is especially useful for topping off walls, forming the refractory cones of oil burners, effecting emergency repairs in the wall proper, etc.

*Pyrometric Cone Equivalent.

Inasmuch as plastic fire brick is essentially unburned fire brick, moulded in place, and burned in the boiler, it is logical to assume that the best results will be obtained in its use when its installation and burning most nearly parallel the manufacture and burning of a good fire brick. It was so assumed in the Navy and installations were made with that object in view.

The material was tightly rammed in place, using the utmost care to avoid both laminations and voids in the material, and to minimize the cracking of the material, due to its inherent shrinkage when dried and burned. It was found that the effect of any laminations in the material could be minimized by building and ramming in horizontal rather than vertical layers; that voids could be prevented by having a sufficient amount of moisture in the material and ramming hard; and that the shrinkage was less if an excess of moisture in the material was guarded against. The cracking was governed to some extent in large installations by "carding" or cutting grooves in the material, the cracks along these grooves being filled with fire clay or plastic after the first firing.

The plastic installation was then air dried, after which it was dried with a slow wood fire placed in the furnace. As soon as convenient thereafter the boiler was lighted off, and the fires worked at a gradually increasing rate up to the maximum, when it was considered the installation was fired and complete. An examination of this method shows that, roughly, the material went through the usual drying, water-smoking, dehydration, and vitrification cycle used in kiln burning fire brick.

The results obtained from the use of plastic fire brick, as outlined above, were not always satisfactory. There were numerous instances of direct complaint, it having been reported that excessive cracking developed, that the wall crumbled, that the face of the wall spalled, or even that the wall just "fell down." A great many more instances of indirect complaint occurred; i.e., certain ships requested certain brands of plastic material—probably because they had seen a successful instance of its use, and had witnessed failures of various other brands.

Due to these unsatisfactory reports the Bureau of Engineering, Navy Department, directed the Fuel Oil Testing Plant, in the

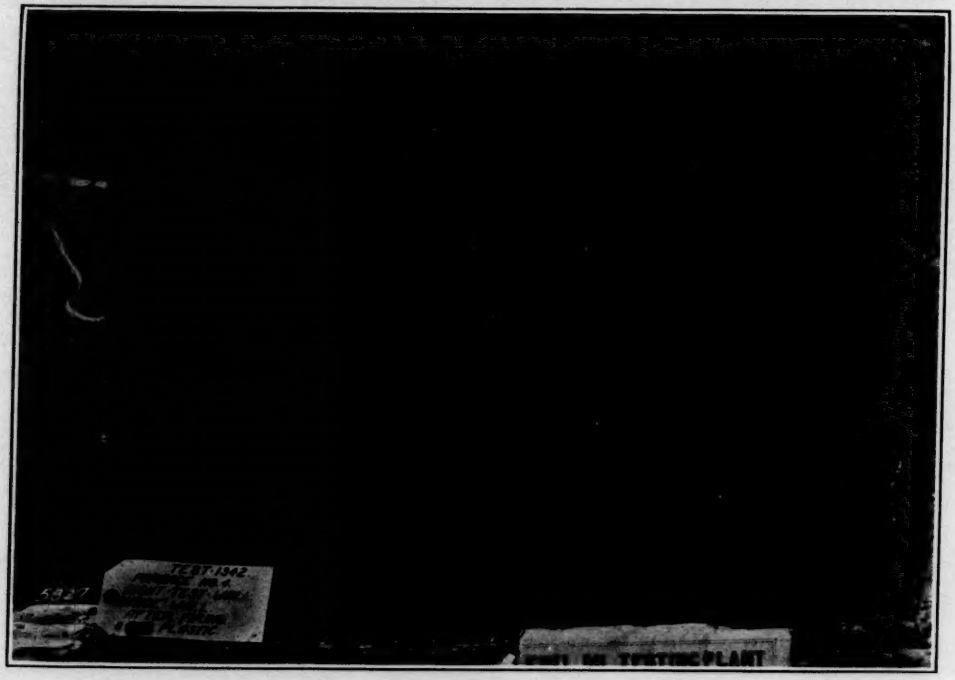


FIGURE 1.

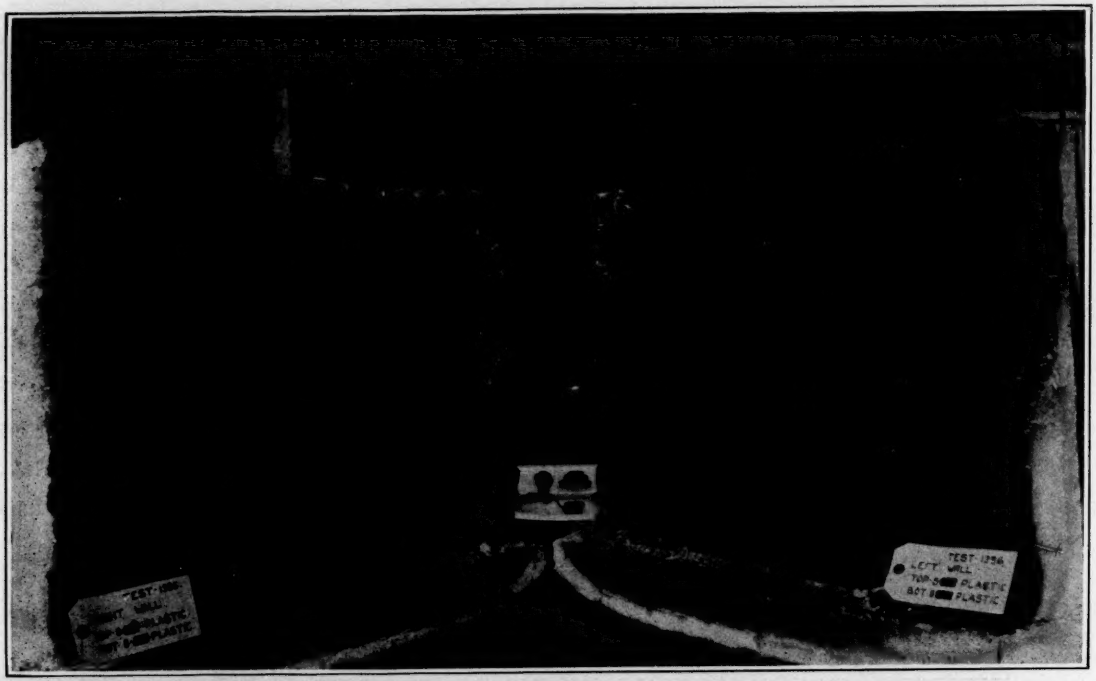


FIGURE 2.

latter part of 1928, to conduct tests designed to better the characteristics of the material supplied, and, if possible, improve the method of installation.

The Fuel Oil Testing Plant had tests in its files on all the materials under question. These tests showed that these materials, when properly rammed, dried and fired would stand furnace temperatures up to 3000 degrees F., with a good resistance to fusion, spalling, excessive cracking, and excessive shrinkage. On these tests, the outside of the test wall was approximately 2000 degrees F. Consequently these tests gave excellent data on the performance of the material over a temperature range of 2000 degrees F. to 3000 degrees F. and over this range, the material, properly worked, had given a hard, dense, strong wall, tending, in some instances only, to be somewhat weak and friable on the 2000 degree F. face. The Fuel Oil Testing Plant was convinced that the material supplied would give satisfactory results if properly installed and fired to the temperatures at which they were tested. It was therefore decided to try to determine, and to duplicate in test furnaces, the conditions that had brought about unsatisfactory results in service.

As a first step, the temperatures actually existing in a typical plastic installation were determined by means of thermocouples installed in the material. This process was simple and of no considerable interest. The results, however, were quite illuminating. It was found that in plastic used for topping off walls in the wake of boiler drums (for example on top of the side walls of an express boiler) or in the plastic refractory cones about the oil burners in the boiler front, the temperatures 1 inch from the furnace face were often as low as 1000 degrees F. Seldom did the temperature at these points exceed 2000 degrees F. (the lowest temperature at which the material had been tested in the test furnace).

This led to a series of test furnaces run at low furnace temperatures, to determine the performance of the material in a low temperature range. In this case the furnace temperature was 2500 degrees F., with an outside wall temperature of approximately 1500 degrees F. The walls were weak and friable, and not properly matured. That is, the temperature in many places

FIGURE 2.

in existing plastic installations is too low to properly mature the material. To obtain a strong wall in those positions, it appeared that the characteristics of the plastics being supplied would have to be radically altered.

The problem was submitted to various refractories manufacturers, and they offered to assist in its solution.

The general opinion seemed to be that a plastic refractory should be used that would fulfill the current high temperature requirements, but which would mature and develop strength over a greater temperature range. With this end in view, various experimental plastics were manufactured and submitted for test.

The experimental plastics submitted were generally stronger than the materials in use, when fired at a low temperature. However, in each case the strength at low temperatures was accompanied by a loss of some desirable characteristic at high temperatures, i.e. at 3000 degrees F. the material developed excessive shrinkage, or showed a poor resistance to cracking, or the material showed a low pyrometric cone equivalent, etc. It soon became apparent that the limitations of refractory materials or the limitations of the present method of their manufacture would not allow us to obtain a material satisfactory in all respects over such a great temperature range. Figure 1 shows a typical wall of material that was quite strong in the lower temperature ranges, after it had been fired to 3000 degrees F. furnace temperature. At the higher temperatures, the material developed spalling cracks, surface cracks on the furnace face, and considerable incipient fusion on the furnace face. An examination of the sag in the top edge of the wall will indicate to some extent its unsatisfactory condition in regard to shrinkage and softening.

At this point it was suggested that some artificial means of obtaining a heat penetration into the wall be employed. The method suggested was to vent the wall, perpendicularly to the furnace face. A test furnace was run at the low temperature of 2000 degrees F., using vent holes varying from 1/8 inch to 1/2 inch diameter. This wall showed such an improvement in strength that it was decided to run further furnaces using the same method. The 1/8 inch hole extending nearly through the wall appeared best for this further testing. The problem of the vented wall appeared to be to con-

FIGURE 3.

FIGURE 4.

firm the indication of increased strength shown by this first wall and to determine whether or not vents materially decrease the insulation value of the wall, and whether or not vents lower the resistance of the wall to high temperature fusion.

A number of furnaces were run using the same materials as were used in the first series, with the walls vented with 1/8 inch holes, spaced approximately 2 inches apart. Without exception the vented walls were somewhat stronger than their corresponding solid walls. In particular, the material near the outer face of the vented wall was considerably less friable than in the case of solid walls.

The following is a typical comparison of temperature drops through vented and unvented 4 1/2 inch walls, the materials in each comparison being the same.

TEMPERATURE DROP.

Furnace Temp. Degrees F.	Vented Wall Degrees F.	Unvent. Wall Degrees F.
2000	475	495
2900	1060	1050
3000	1040	1120

The difference in temperature drop between the vented and unvented wall was negligible, and within the accuracy of the instruments employed in this testing.

To determine the effect of the vents on plastic performance at high temperatures, a furnace was built, using four different brands of material, and fired at 3000 degrees F. for 100 hours. The materials selected had a P.C.E. varying from the highest found on the Navy Approved List of Materials (Cone 33) to a material of Cone 30, one cone less than is required by current specifications. Figure 2 shows this furnace after firing. The walls are strong and dense. No undue fusion is apparent excepting on the face of the low cone material where the fusion was sufficient to close some of the vents.

Up to this time the Fuel Oil Testing Plant had done little but investigate the cause and remedy of weak and friable walls. It now undertook to investigate the cause of excessive spalling of

plastic walls. A consideration of the cycle used in kiln-burning refractories indicated that this condition was probably due either to (*a*) excessive moisture in the wall when first fired, or at any time thereafter or (*b*) no proper means provided to allow moisture to escape from the wall when it was fired. It was of course assumed that the wall had been properly rammed.

Both conditions were duplicated in the test furnaces. For the first condition, a wall was rammed and dried by air and a very slow fire until the furnace face was apparently bone dry. The outside face of the wall was still slightly damp. For the second case, the wall was rammed and the furnace face trowled. This is often done in the field by poorly instructed workmen to give the job a somewhat better appearance. This second wall was dried until it was apparently bone dry. When fired, both walls gave the same result. In the first case, the large amount of steam generated in the wall interior ruptured the wall and caused the furnace face of the wall to fall off in a 2 inch layer, parallel to the furnace face. In the second case, a small amount of steam generated in the interior of the wall, also caused rupture, due to the closed pores, and "case hardening" of the furnace face of the materials. In both cases the failures occurred before the furnace had reached an operating temperature.

The venting of the wall, which had so improved the strength of the wall, also offered a solution for the problem of elimination of wall moisture. This solution was tried in a test furnace under extreme conditions. The walls were rammed, vented with 1/8 inch holes as has been described and set in place as soon as they were dry enough to support themselves. The wall was dried on the furnace face by means of a portable blower until it felt dry to the hand. No drying by a gentle fire, or any approach to that method was used. The furnace was then rapidly fired to a furnace temperature of 3000 degrees F. where it was held for 30 hours.

Of the complaints made, the direct ones concerning excessive cracking, and the indirect ones embodied in requests for proprietary material remained to be investigated.

Cracking of plastic fire brick is due primarily to the shrinkage of the refractory material when it is dried and burned. Although

the amount of shrinkage varies somewhat in different materials, it is inherent in all of them. These shrinkage cracks in plastic walls are part of the price paid to eliminate numerous and expensive shapes and to insure having always available the material with which to maintain the furnace installations. It is essential that this characteristic of plastic refractories be understood before complaints on shrinkage cracking are made. It is doubtful if such is the case.

Although shrinkage cracking in large plastic areas is an inherent defect in the material, the effect of this cracking on the life of the furnace lining may be minimized by a proper understanding and control of the cracking. In general, the cracks occur across the areas having the least cross-section. The effect of this is to divide the plastic wall, particularly in the furnace front (in wake of the oil burners) into a number of refractory blocks, *each of which must be properly supported by an ample arrangement of anchor bolts*. For example, in 1928 there was delivered to the Fuel Oil Testing Plant an Express type boiler, a duplicate of the boiler installed on the U.S.S. *Salt Lake City* class of cruisers. As delivered, about 12 anchor bolts were provided in the plastic area forming the furnace front of 13 fuel oil burners. The first refractory front installed in this boiler had a very short life. After the initial cracking, large segments of the material became loose, came away from the front plate and finally commenced falling into the furnace. A second installation gave similar poor results. Figure 3 shows this second wall after various vain attempts at satisfactory repairs. The loose sections about the center burner are very apparent. On board ship with the vibrations to which the whole ship in general, and boiler fronts in particular, are subject, the front of this boiler would have given even a shorter life.

This boiler offered an opportunity to test certain measures for minimizing the effect of plastic cracking, as well as to give a comparative service test of various brands of acceptable plastic materials. The number of anchor bolts in the furnace front plate was quadrupled, giving about 4 per burner opening. A front was installed using the old solid ram, wood fire drying method, and eight different brands of plastic materials. Of these, seven were on the Navy Approved List of Materials, while one material, of

the air-setting variety was not in the acceptable class. The "Acceptable List" materials were representative of all refractory producing districts in the country, the Eastern, Middle West, and Western fields being represented. After the first steaming period of the boiler, the air setting material was fused badly, and was replaced by an acceptable plastic, rammed using the venting method previously described.

Figure 4 shows this installation after nine months intermittent service at all steaming rates. The entire wall was strong, and was firmly held to the front plate. From a surface inspection, the various brands of material could not be distinguished. The shrinkage cracks were easily filled, using only a fire clay wash. The three vented cones, installed subsequent to the failure of air-setting material, and fired without any special drying, are clearly shown. These three cones were in excellent condition.

The work on this boiler front indicated that many of the failures in service due to "crumbling" or "falling down" of the wall (using the terms in which the complaints are usually made) are probably due to a simple lack of proper anchor bolting of the material when installed. It was further indicated that, in general, proprietary requisitions for plastic fire brick material are unjustified. Any material on the Acceptable List will give results comparable to the other materials, if it is properly installed and properly fired.

For checking the installation of a "vented wall" in service, two opportunities presented themselves. The first was the Sectional Express boiler, delivered to the Fuel Oil Testing Plant in April 1929. When this boiler was received, three anchor bolts were installed in a plastic area covering five burner cones, a condition that would probably lead to the same failure of sections of the wall, as had been experienced in the *Salt Lake City* type express boiler. The number of anchor bolts was increased to 14. The materials (one from an Eastern field and one from a Middle Western) were rammed and vented nearly through the wall, perpendicularly to the plastic furnace face, using 1/8 inch rod for venting. No artificial means of drying was used. The wall was allowed to air-dry, until the furnace surface appeared dry to the touch, and then the boiler was fired. Figure 5 shows this front after six months intermittent steaming at all rates while the boiler

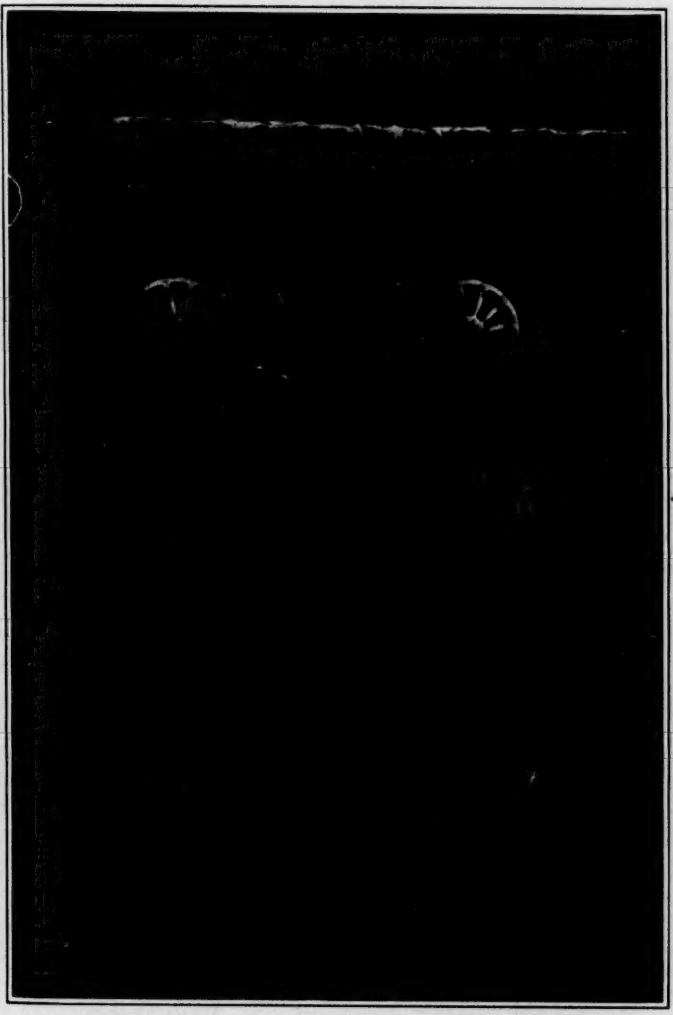


FIGURE 5.

was being tested. The cones were given one fire clay wash after approximately 100 hours steaming. The wall is still in place and is strong, and firm, with only the usual shrinkage cracking.

The foregoing summarizes briefly the results of a year of tests on plastic fire brick materials. Some of these results only confirmed the previous experience of the Fuel Oil Testing Plant on the subject, while others, particularly the method of venting plastic walls, gave some new and interesting information. Following is a resumé of all determinations made.

(*a*) The Fuel Oil Testing Plant was unable to obtain a plastic fire brick material that developed strength over a materially greater temperature range than do those on the current Approved List without sacrificing desirable high temperature characteristics.

(*b*) Venting a plastic wall, as described, adds to the strength of the interior of the wall.

(*c*) Venting a plastic wall, as described, does not materially decrease the insulation value of the wall.

(*d*) Venting a plastic wall, as described, does not cause a loss of high temperature characteristics of the material.

(*e*) Failure of plastic walls in service may easily be caused from firing a damp wall, a wall with a troweled surface, or a wall that is insufficiently supported by anchor bolts.

(*f*) Venting a plastic wall allows it to be fired from an air dried condition, without recourse to wood fires for drying, etc.

(*g*) Venting a plastic wall does not increase its liability to cracking.

(*h*) A vented plastic wall may be fired with a slight moisture content with no harmful effects.

(*i*) Plastic materials on the Navy Approved List of Materials, under the same conditions, give substantially comparable results in service. Proprietary purchases are usually not warranted.

(*j*) The number of anchor bolts employed in a plastic wall of a Naval boiler should be equal to or exceed the number used in the same area when using moulded shapes. Anchor bolts give best results when staggered both in depth and in line.

(*k*) Less care is required in drying and firing a vented wall than is required in drying and firing a solid wall.

(*l*) A plastic wall may also fail due to too great an interval occurring between drying and firing the wall. It is improbable that any failures in service are due to this cause.

The Fuel Oil Testing Plant plans, as opportunity offers, to continue experimentation on the selection and working of plastic materials. However, it is considered that the proof is conclusive that the method of venting a plastic wall, as has been described, and firing from an air dried condition, so increases the reliability of the installation and so simplifies the question as to when a wall is dry enough to fire, that it should be used in preference to the present method of ramming the plastic solid, and drying by means of a wood fire, or other artificial means.

AUTOMATIC STEERING TESTS.

By N. Minorsky, Member.

INTRODUCTION.

This paper describes a series of tests of a new automatic steering apparatus authorized by the Bureau of Construction and Repair of the U. S. Navy and carried out by the author on the U. S. S. *New Mexico*. It was thought that the publication of the results of these tests might be of general interest in view of the absence of any engineering publication dealing with the underlying principles of the design and performance of automatic steering apparatus. The publications available are mostly of a commercial character without any engineering analysis of the dynamical phenomena involved.

The object of this paper is to analyze the fundamental dynamical problem as well as to describe certain attempts made towards its solution, rather than to elaborate on a description of details which today are of small interest in view of the rapid advance of technique during the past six years. In order to make the description more definite, some of these details are mentioned in a general manner.

The *New Mexico* tests took place at a time when the commercial automatic steering apparatus just began to appear on the market (1923) and when, consequently, no preceding engineering experience or information was available. Furthermore, the size of the vessel as well as the accuracy of steering which was finally obtained after a series of efforts might be of interest as an example of the control of directional stabilization of masses possessing great moments of inertia and very small initial damping.

This kind of problem is characteristic of modern ships irrespective of whether one has to do with the problem of automatic steering or with that of the anti-rolling stabilization of ships or

with particular problems, for example stabilization of certain parts of ships such as landing decks, guns, torpedo tubes, searchlights and the like.

For more information as to the underlying principles of the continuous dynamical control used in this apparatus, the reader is referred to an earlier publication of the writer.* In the present article are mentioned only such theoretical points which are necessary for a complete understanding of the *New Mexico* installation.

The description of the tests is arranged in a chronological order in order to present as clearly as possible the gradual evolution of the automatic steering apparatus after the first tests at sea and to show in this manner the scope of the whole problem with its inherent difficulties and limitations. The orders of magnitude of various dynamical quantities entering into control were unknown at the beginning. They became known only after the first test in Puget Sound and the subsequent adjustments were guided mainly by general theoretical considerations which at all times were found to be very valuable as a guide towards the final adjustment.

It appears quite well established that the method of small oscillations, forming the foundation of the theory on which the apparatus was designed, gives a correct insight into the dynamics of automatic steering.

Throughout the whole period of the tests, the ship was acting as Flagship of the Battleship Divisions and most of the time was leading the formation. For this reason, no continuity of effort in carrying out the required adjustments and observations could be maintained throughout the whole period when the writer remained on board the *New Mexico*. These tests, therefore, were scattered over a considerable period of time and no systematic study of all the various factors was possible under the conditions. Only on comparatively rare occasions, when the ship could conveniently fall out of formation, was it possible to carry out the actual work of adjustments, without being bothered by the zigzagging inevitable in such cases.

In spite of these difficulties, owing to the kind interest in this device on the part of the Commanding Officer, Captain Yates

* N. Minorsky, "Directional Stability of Automatically Steered Bodies." JOURNAL OF AM. SOC. OF NAVAL ENGINEERS, MAY, 1922.

Stirling, Jr., U. S. N., the necessary amount of automatic steering adjustments could be obtained and the device could thus be gradually perfected.

The main credit for the strenuous period of gradual adjustments described in Part II is due without doubt to the electrical officer of the ship, Lieutenant J. R. Redman, U. S. N., as well as to the men assigned by him for this purpose; they really have spared no effort to bring the automatic steering performance finally to the state of very great accuracy described in section 12.

Part III deals with the conclusions derived from the *New Mexico* tests and establishes a general definition of a dynamically correct phase of the control action. This principle was utilized in the *New Mexico* apparatus and its gradual perfection after the first tests at sea.

Finally some potential possibilities of this new method of control which, for the reasons which will be apparent later, is designated as *parametric*, are presented, as a personal opinion of the writer, in conjunction with the probable evolution of naval warfare in the years to come.

UNDERLYING PRINCIPLES OF THE AUTOMATIC STEERING CONTROL.

The *New Mexico* steering gear was designed in accordance with the principles set forth in the article previously mentioned. In the terminology of that article the *New Mexico* automatic steering control belonged to the steering control of the first class, that is of the type which is generally defined by the relation:

$$\rho = m\alpha + n\frac{d\alpha}{dt} + p\frac{d^2\alpha}{dt^2} = \rho_1 + \rho_2 + \rho_3$$

where ρ is the rudder angle,

α the angle of deviation of the ship from its set course,

$\frac{d\alpha}{dt}$, $\frac{d^2\alpha}{dt_2}$ the angular velocity and acceleration of the yaw, respectively.

In the original installation the angular acceleration control was absent; later on, after the first trials, the accelerational feature was added, as will be more fully explained later.

As regards ρ_1, ρ_2, ρ_3, they may be considered as *component rudder angles,* defined by relations $\rho_1 = m\alpha$, $\rho_2 = n \frac{d\alpha}{dt}$ and $\rho_3 = p \frac{d^2\alpha}{dt^2}$. The introduction of these component rudder angles is rather a mathematical convenience for the analysis of the performance. In fact, within the limit of small motions with which we are concerned here, the principle of superposition is applicable because the differential equations are linear, at least in the first approximation. In other words, everything happens as if the ship were controlled by two independent rudders (assuming here $p = o$). of which one is controlled by the angle α (i.e. $\rho_1 = m\alpha$) and the other by the angular velocity $\frac{d\alpha}{dt}$ $\left(\text{i.e. } \rho_2 = n \frac{d\alpha}{dt}\right)$.

For the theoretical foundations of this method, the reader is referred to the earlier article; it is sufficient to mention here that the determination of coefficients m and n constitutes the main part of the adjustments to be carried on board ship. In fact, nothing a priori indicates the proper magnitude for these coefficients, which depend on such factors as the gradual building up of pressures on various parts of the hull when the rudder has just been moved, on the functional relation between the angular velocity of yaw and the resistance of the water to yawing and similar unknown factors.

But whereas the correct magnitude of these coefficients is unknown initially, the variation of these coefficients is readily accomplished by suitable change in ratios of the corresponding transmissions, adjustment of electrical circuits, etc.; namely, the coefficient m depends on all ratios of transmission between the gyroscopic repeater compass and the control shaft of the Waterbury hydraulic gear moving the rudder; in a similar manner the coefficient n depends on all ratios between the, so-called, gyrometer * (see Section 5), otherwise angular velocity responsive instrument and the same control shaft.

The convenient place for adjusting the values of these coefficients was on the instrument side of the control equipment.

* U. S. Patent No. 1372184.

STEERING GEAR.

The choice of the U. S. S. *New Mexico* for automatic steering tests was largely due to its electro hydraulic steering gear, which, in the opinion of the Bureau was particularly well adapted for continuous regulation in response to the continuous control produced by the automatic steering apparatus. It is clear that with the steering gear controlled in a discontinuous manner (e.g. by contactors or an equivalent arrangement), the advantages of the continuous control produced by the instruments could not be properly ascertained.

An electro hydraulic steering gear,† from this viewpoint, has characteristics particularly suitable for the contemplated problem, insofar as it permits establishing a series of continuous control actions from the instruments down to the rudder.

These theoretical considerations, later on, found their complete confirmation in the results of trials of the automatic steering apparatus.

PILOT MOTOR, BRAKE, CLUTCH.

The connection of the automatic steering instrument with the Waterbury hydraulic gear, actuating the rudder, was established through an electromagnetic clutch at the control shaft of the hydraulic gear.

Figure 1 shows the pilot motor of the automatic control. It is designated in the following as *automatic pilot motor* in order to distinguish it from the pilot motor of the manual control; the last mentioned motor is designated in the following as *manual pilot motor*.

The automatic pilot motor had the following characteristics: 2.5 H.P., 230 volts, 1000 R.P.M. rated speed; it was mounted on a special bed plate with a series disc brake and clutch as shown on Figure 1. The trick wheel A normally was placed on the control shaft of the Waterbury gear and one of the spur gears B was secured on the end C of the shaft. The connection of the auto-

† The reader is referred in this connection to an article by Commander H. S. Howard (CC) U. S. N., entitled "Hydraulic Steering Gears," JOURNAL OF AM. SOC. OF NAVAL ENGINEERS, May, 1922.

matic pilot motor with the control shaft of the Waterbury gear was thus obtained by means of the engagement of the gears B and A.

In accordance with the instructions of the Bureau, the manual pilot motor must at all times remain in a permanent mechanical connection with the control shaft of the gear. To obtain this result, the brake of the manual pilot motor was provided with an additional shunt coil, whose duty was to release this brake when the automatic steering was taking place. Under those circumstances, the armature of the manual pilot motor was able to rotate idly, as a flywheel, when the automatic steering apparatus was controlling the rudder.

This particular arrangement turned out to be very convenient during the period of adjustments at sea, when the ship was steered by the quartermaster. By running the automatic steering device disconnected from the rudder by the open clutch, it was possible to compare its controlling action (part D of the clutch) with that produced by the quartermaster (part E of the clutch).

AUTOMATIC STEERING INSTRUMENT.

A general view of the automatic steering instrument is given on Figure 2, representing the instrument on the turn table of the Steward Davit & Equipment Corporation, Hudson, N. Y., where the instrument was built and tested prior to its shipment to the Pacific Coast.

The instrument contains three parts:

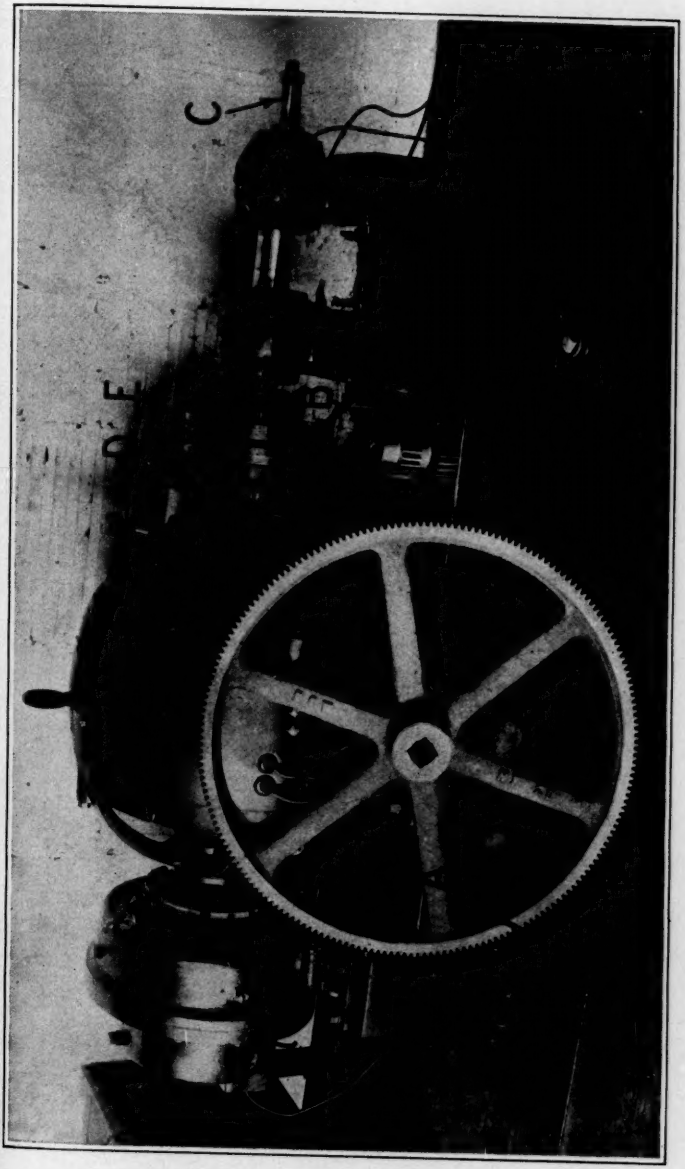
(*a*) Compass control.

(*b*) Gyrometer control.

(*c*) Combination of both controlling actions through a differential gear, operating the potentiometer circuit.

The compass control was comparatively simple; it contained an ordinary 10′ step-by-step motor from the standard Sperry gyroscopic compass, already available on the ship; by means of an ordinary follow-up motor the rotation of the repeater compass motor through a certain angle was accompanied by the rotation of the follow-up motor through a proportional number of revolutions.

The gyrometer control had two gyroscopes (of which one is seen on Figure 2 through the dust-proof gauze). The gyroscopes



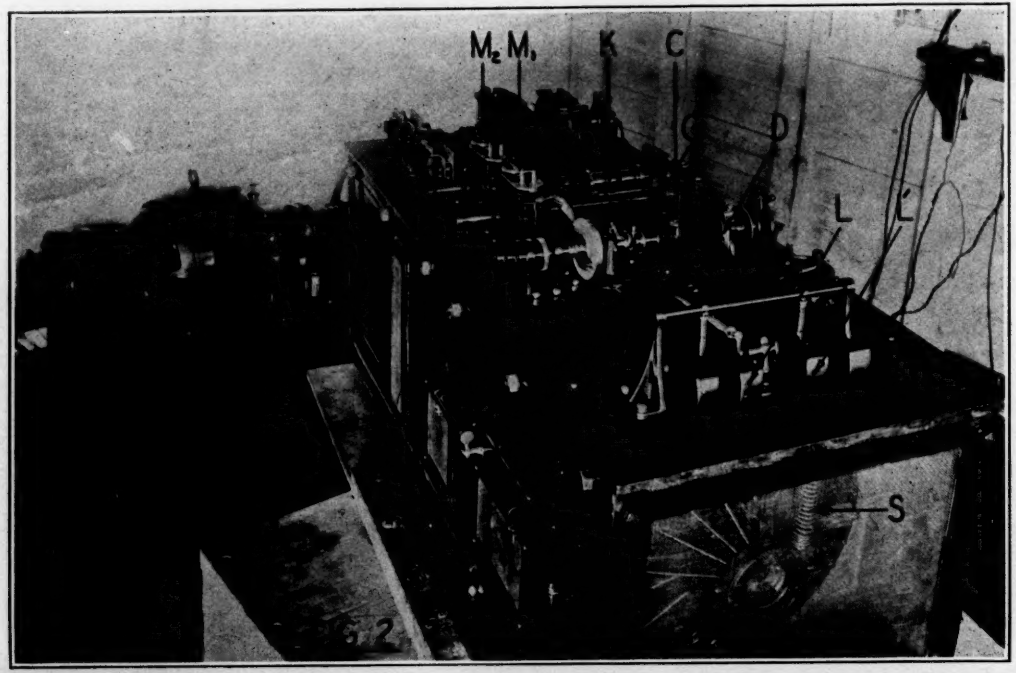
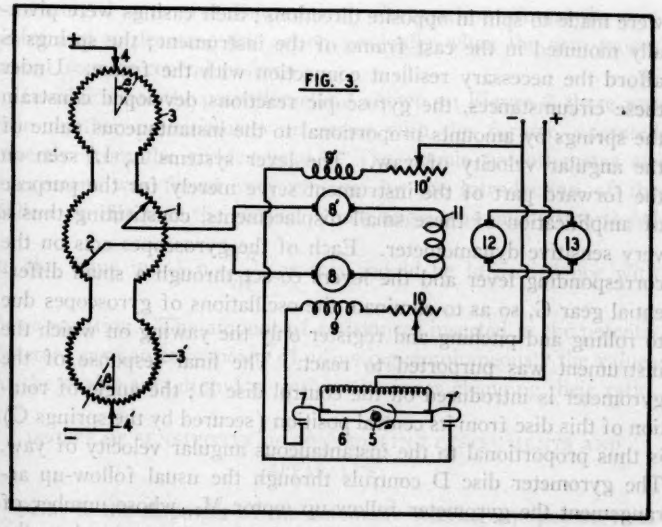


Fig. 2

FIG. 3.


+
4
3
1
2
5
4
−
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11
12
13
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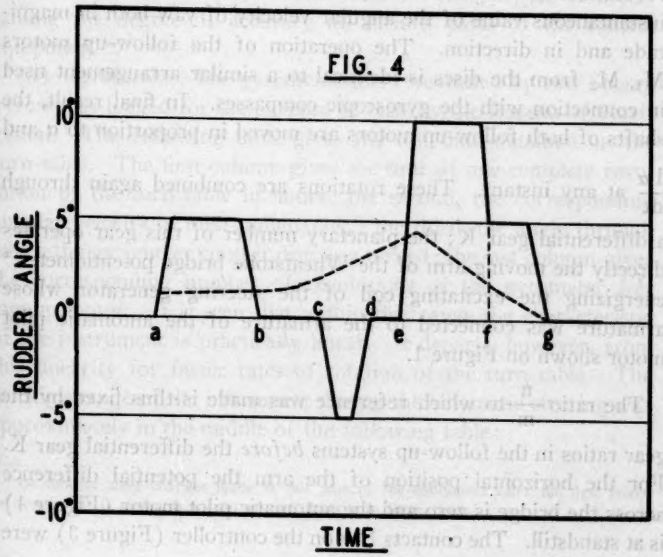
7
6
5

FIG. 4


10°
5°
0°
-5°
-10°
RUDDER ANGLE
a
b
c
d
e
f
g
TIME

were made to spin in opposite directions; their casings were pivotally mounted in the cast frame of the instrument; the springs S afford the necessary resilient connection with the frame. Under these circumstances, the gyroscopic reactions developed constrain the springs by amounts proportional to the instantaneous value of the angular velocity of yaw. The lever systems L, L′ seen on the forward part of the instrument serve merely for the purpose of amplification of these small displacements, constituting thus a very sensitive dynamometer. Each of the gyroscopes acts on the corresponding lever and the levers co-act through a small differential gear G, so as to eliminate the oscillations of gyroscopes due to rolling and pitching and register only the yawing on which the instrument was purported to react. The final response of the gyrometer is introduced on the control disc D; the angle of rotation of this disc from its central position (secured by the springs C) is thus proportional to the instantaneous angular velocity of yaw. The gyrometer disc D controls through the usual follow-up arrangement the gyrometer follow-up motor M_1, whose number of revolutions from its middle position is thus proportional to the instantaneous value of the angular velocity of yaw both in magnitude and in direction. The operation of the follow-up motors M_1, M_2 from the discs is identical to a similar arrangement used in connection with the gyroscopic compasses. In final result, the shafts of both follow-up motors are moved in proportion to α and $\frac{d\alpha}{dt}$ at any instant. These rotations are combined again through a differential gear K; the planetary number of this gear operates directly the moving arm of the Wheatstone bridge potentiometer * energizing the excitating coil of the steering generator whose armature was connected to the armature of the automatic pilot motor shown on Figure 1.

The ratio $\frac{n}{m}$ to which reference was made is thus fixed by the gear ratios in the follow-up systems *before* the differential gear K. For the horizontal position of the arm the potential difference across the bridge is zero and the automatic pilot motor (Figure 1) is at standstill. The contacts (2) on the controller (Figure 3) were

* Described in connection with Figure 3.

so arranged as to give comparatively sharp action around the zero point and to decrease this action gradually when the arm moves away from its central position.

In addition to the potentiometer 1 shown on Figure 3, there was also another potentiometer in series with the former, operated directly from the rudder head.* The double potentiometer arrangement was found very efficient by the introduction of the follow-up effect on the rudder, whose angle was thus proportional to the value $m\alpha + n\frac{d\alpha}{dt}$ as this should be in accordance with basic equation. The amount of resistances inserted in the potentiometers gave an easy means of adjusting simultaneously the values of the coefficients m and n, without however changing their ratio.

DEGREE OF SENSITIVITY OF CONTROLLING INSTRUMENTS AND APPARATUS.

Prior to the shipment of the automatic steering equipment, tests were made in the shop on a turn table for the purpose of ascertaining the degree of accuracy of various parts of the control equipment.

(*a*) *Gyrometer*—The gyrometer was constituted by two Sperry gyroscopes loaned for the purpose of tests by the Bureau of Navigation. The following table gives the test data obtained on the turn table. The first column gives the time of one complete revolution of the turn table in hours; the second, the corresponding angular velocity in milli-radians/sec.; the third, the angle through which the gyrometer contact disc was turned; the last column gives the corresponding number of revolutions of the gyrometer follow-up motor. It is seen that within this range the characteristic of the instrument is practically linear. It departs, however, from this linearity for faster rates of rotation of the turn table. The working range of the instrument during the automatic steering was approximately in the middle of the following table:

* In view of some changes made in this part of the equipment after the first trials, its description is given in a later section describing these trials.

Hours	M. Rad. Sec.	Degrees	Revs.
.5	3.5	44.0	77
1.0	1.75	22.5	39
2.0	.875	11.25	19
4.0	.437	5.5	10
8.0	.219	2.75	5

The overall ratio of transmission between the gyrometer control disc and the corresponding number of revolutions of the follow-up motor was thus 640 : 1.

It is interesting to note that on the last point the turn table was rotating at a rate only 1.5 times greater than that at which an hour hand of a clock rotates; to an angular motion of this rate the follow-up motor was already responding by 5 revolutions which was sufficient to start the controlling action.

(*b*) *Compass*—An ordinary Sperry's 10′ repeater compass motor was used in the control; the ratio between the number of revolutions of the compass steering motor and control disc fixed on the shaft of the step-by-step motor was 320 : 1.

(*c*) *Automatic Pilot Motor*—The maximum speed of the control shaft of the Waterbury hydraulic pump in the existing manual pilot motor is about 160 R.P.M.

The automatic pilot motor was designed to operate the control shaft of the hydraulic gear at much lower speed. The following data relate to the maximum and minimum speeds of the motor generator set and for each case the maximum and minimum speeds of the automatic steering motor are given depending on the position of the potentiometer contact.

(*a*) *Maximum Speed of M. G. Set*—

Maximum speed of control shaft—150 R.P.M.

Minimum speed of control shaft—62 R.P.M.

(*b*) *Minimum Speed of M. G. Set*—

Maximum speed of control shaft—17.5 R.P.M.

Minimum speed of control shaft—4.5 R.P.M.

It is seen that the speed range of the control shaft was considerably below the usual range of speed of that shaft when operated by the quartermaster. The important result of this condition, as

far as the automatic steering was concerned, was almost noiseless, so to say micrometric operation of the rudder by very small amount but practically all the time; whereas in the case of manual operation by the helmsman the steering gear is operated with great noise, by large amounts, from time to time.

CHANGE FROM MANUAL STEERING TO AUTOMATIC AND VICE VERSA.

No additional disconnecting switches were allowed and the changes from manual steering to automatic and vice versa must be produced from the control handle of the steering controller. To meet this requirement the following arrangement was made: the control handle of the steering controller was made hinged; for manual steering this handle was normally in horizontal position, except when it was facing a gap in the middle, which was normally bridged. When the automatic steering was desired, the quartermaster had to remove the bridge from the gap and let the handle go to its central position; owing to the weight attached, the handle is pulled into this gap, rotating about the hinge, which closes an auxiliary contact of the *interlocking circuit*.

The whole operation of the transfer is accomplished by means of this interlocking circuit. In view of the fact that the latter does not present any special interest being formed by common industrial relays and contactors, it is sufficient to mention only about the sequence of operations produced by that circuit:

(*a*) Electromagnetic clutch (Figure 1) connects the automatic pilot motor to the control shaft of the hydraulic gear.

(*b*) At the same time an additional shunt brake on the manual pilot motor releases its brake, whereby the armature of this motor is allowed to run idly as a flywheel.

(*c*) Follow-up motors are energized and begin to follow the sensitive elements of the compass and the gyrometer, respectively.

(*d*) Gyroscopic compass is connected through an additional clutch with the control disc of the compass follow-up motor. Owing to this arrangement, the instantaneous bearing of the ship at the instant of closing the interlocking circuit is introduced as the zero point about which the automatic steering begins to function.

For an opposite operation (i.e. for a change from automatic to manual steering) the quartermaster had to remove the handle of

the steering controller from the gap which disconnects the interlocking circuit; the sequence of the control actions was as follows:

(*e*) The electromagnetic clutch is de-energized, which disconnects the automatic pilot motor from the control shaft of the hydraulic gear.

(*f*) The shunt coil of the manual pilot motor is de-energized so that this motor becomes released only by its series brake operated by the helmsman through the steering controller.

(*g*) Both compass and gyrometer follow-up motors, by means of a special centralizing circuit, are brought to the central position ready for the next connection with automatic steering instrument.

PART II.

TESTS OF THE AUTOMATIC STEERING APPARATUS.

TEST IN PUGET SOUND.

The first trial was made while the ship was proceeding in Puget Sound from Bremerton en route for Bellingham. This test was of very short duration, since no good steering could be obtained; the rudder was moved by a large amount and the general impression was that a very considerable undamped yaw was maintained, owing to a wrong phase of the rudder action. The test hardly lasted one minute, after which the manual steering was resumed.

An analysis of the performance indicated that:

(1) The damping control of the gyrometer was not felt sufficiently.

(2) The rudder was moving too fast.

It was decided that the first condition could be improved by increasing the gear ratio on the transmission of the gyrometer control and by decreasing the corresponding ratio on the compass control. This work was carried out by the mechanical repair shop on board ship.

As regards the second condition, considerable predominance was given to the potentiometer control derived from the rudder, which increased the follow-up character of the rudder control and considerably reduced its speed, all other conditions remaining the same.

TESTS EN ROUTE FOR SAN FRANCISCO.

During that period, two tests of a comparatively short duration (between 2 and 3 minutes) were produced. The first test was made soon after the ship left Bellingham. During the first swing, the ship moved away by about 3 degrees; on opposite swing it reached about 2.5 degrees; and on the following swings approached the value of about 2.0 degrees. It seemed thus as if an undamped yaw had a tendency to be stabilized at that amplitude of 2.0 degrees. Further observations were interrupted by the passage again to the manual steering. During this test, the ship was leading the formation and any further experimentation with steering was found objectionable.

It was decided that for another two or three days, in view of contemplated tactical evolutions, no further experiments should take place.

It was clear from the preceding test that although both the phase and intensity of the control action were considerably improved as compared with the Puget Sound test, the phase of the control was not sufficiently advanced to exert adequate "meeting" and "easing off" actions on the rudder. Theoretically, two methods of attack were available:

(1) To increase the intensity of the gyrometer control.

(2) To introduce the accelerational control which, thus far, was left out entirely.

It was found difficult to increase the ratio in the gyrometer control without any considerable changes in the design of this apparatus. As regards the second alternative, while the introduction of a continuously acting accelerational control was out of the question, in view of the considerable amount of work involved, it was found that a discontinuous accelerational control acting on the crest value of angular velocity of yawing could be done with comparatively simple means. A control of this type while being devoid of the continuity in the angular acceleration (i.e. second time derivative $\frac{d^2\alpha}{dt^2}$ of angular motion) was expected to improve considerably the phase as well as the intensity of the angular velocity control.

For this purpose on the shaft of the gyrometer follow-up motor was placed, with a certain amount of friction, a lever,* bearing two contacts, one on each end. These contacts were short circuiting certain resistors in the controlling potentiometer. As long as the gyrometer follow-up shaft was turning in the same direction, that is as long as the angular velocity of yawing was steadily increasing, the same contact remained closed. As soon as the angular velocity, having reached a maximum, begins to decrease, the gyrometer follow-up shaft begins to rotate in the opposite direction and the lever closes another contact. Thus the change of contacts occurred strictly on the maximum points of angular velocity, or, which is clearly equivalent, on the zero points of the angular acceleration.†

This particular control feature had a very simple dynamical significance. In fact, the angular acceleration of the ship, in virtue of the Postulate of Dynamics, is equal at any instant to the difference between the disturbing torque (due to various factors such as waves, difference in propelling thrust of the engines, wind, etc.) tending to produce the yaw and the restoring torque of the rudder. The instant when the angular acceleration is zero characterizes, therefore, the equality between the disturbing and the restoring moments. The change of the controlling contacts occurring at this particular point was giving thus the earliest response of the control system to the tendency of the system to move under the influence of the resultant moment.

A temporary arrangement was designed and built by the ship's crew; in view of its resemblance in operation to the movements of a grasshopper preparing itself for a jump, it was christened by the crew "grasshopper contact" or simply "grasshopper."

The "grasshopper" was completed two days before the arrival of the Fleet at San Francisco. Observation of the automatic steering motor's performance, while being disconnected from the rudder (as explained at the end of Section 4) showed this time that the phase of the control action was almost in synchronism with the manual control. The next day permission for connection with the

* A more complete description of this arrangement is given in the following section.

† From the analytical definition of derivatives to the maximum value of the first time derivatives ($d\alpha/dt$) corresponds to the zero point of the second ($d^2\alpha/dt^2$).

steering gear was given and on the first swing the ship reached only about 1.5 degrees, and on the following swings stabilized itself roughly on 3/4 degree of undamped residual yaw (i.e. from 4 to 5 jumps of the 10′ gyrocompass repeater motor).

In view of the very material success obtained by means of the grasshopper arrangement, it was decided that a regular apparatus incorporating this feature be designed and built by the Mare Island Navy Yard.

CHANGES MADE AT THE MARE ISLAND NAVY YARD.

In the middle of September, Mare Island Navy Yard had completed the acceleration-control attachment which superseded the original "grasshopper" arrangement. The field control circuit of the automatic steering apparatus is shown with this final modification on Figure 3 in which the parts 6, 7, 7′, 8, 8′, 9, 9′, 10, 10′ were manufactured by the Mare Island Navy Yard.

In Figure 3:

1 represents the main potentiometer.

2 is the circuit fastened to the hand of the potentiometer, operated from the planetary shaft of the differential gear K (Figure 2).

3 is the follow-up potentiometer operated from the motion of the rudder displacing contacts 4, 4′ from the position shown into the position indicated by dotted lines under the angle β to the original position.

5 is the shaft of the gyrometer follow-up motor (M_1 of Figure 2).

6 is a lever arm made of insulating material and mounted on the shaft 5 with a certain amount of friction.

7 7′ are the contacts; one of them was generally closed, depending on the direction of rotation of shaft 5.

8 8′ are two armatures of a small (1/8 H.P.) booster set, inserted in series with the potentiometer circuit.

9 9′ are the field coils of the boosters.

10 10′—series rheostats.

11 the field of the steering generator.

12 the armature of the steering generator.

13 The armature of the automatic pilot motor.

The operation of this control circuit is as follows: The arm 2 is rotated from the follow-up motors (of the compass and of the gyrometer) and the angle α' of its rotation from the neutral position is given by the relation $\alpha' = m'\alpha + n'\frac{d\alpha}{dt}$. This introduces a potential difference across the field 11 of the generator 12 and starts the automatic pilot motor 13.

As to the accelerational control, the performance can be described as follows: as long as the angular velocity varies in the same direction (i.e. as long as the angular acceleration has the same sign), the shaft 5 of the gyrometer follow-up motor rotates in the same direction and one of the contacts (e.g. 7′ for the direction of rotation shown by the arrow ↘) is closed. This energized the field 9′ and inserts in the control circuit an additional electromotive force developed in the armature 8′. During this period, the armature 8 is idle.

As soon as the angular velocity reaches its maximum (i.e. as soon as the angular acceleration changes its sign passing instantaneously through the zero value), the rotation of the shaft 5 reverses and this reverses the functions of the armatures 8 and 8′, namely armature 8 becomes active and 8′ idle. The amount of this additional accelerational effect is controlled by the resistors 10 and 10′.

FINAL ADJUSTMENTS AT SAN PEDRO.

On the 18th of September, the installation of the parts furnished by the Mare Island Navy Yard was completed and the preliminary work of adjustments began at once. The ship was under way most every day and the adjustment of the apparatus, by comparing its performance to that of the manual steering, as explained in Section 4, was very convenient. Under the circumstances, the work of adjustments at this stage consisted of finding out the best possible distribution of resistors between the contacts of potentiometers 1, 3 and 3′ as well as in gradual adjustments of resistors 10 and 10′ fixing the magnitude of accelerational control. Already after the first day's work it was observed that the steering device not only reached the state of perfection of the manual steering but began gradually to surpass it by a steadily increasing margin. The

phase of the control action was gradually shifted ahead of that exerted by the quartermaster, at least by 3–4 and sometimes 5–6 seconds. If initially the comparison between the actual manual steering and the performance of the automatic device served to ascertain whether the automatic steering apparatus was "right" or "wrong," now the roles have changed and the observation of the simultaneous performance of both manual and automatic methods was giving now an unfailing indication as to whether or not the quartermaster was right or wrong. During this time a series of observations was made by a number of observers marking the actual angle of the rudder (manual control) and approximately calculated values of the rudder angle if the automatic pilot motor had been connected to the rudder; the scale for this last curve was determined approximately from the actual automatic steering at a later time. These diagrams, in spite of their very crude approximation, were very helpful during this period of adjustment. A typical diagram of this kind is shown on Figure 4 where the trapezoidal curve shows the actual manual and the continuous curve the automatic (not connected to the rudder) controls, plotted against the time.

The rudder angles of the manual control were multiple of 5 degrees, due to the arrangement of the helm indicator, which did not allow the helmsman to ascertain any intermediate angles. No such restriction existed regarding the automatic control. The diagram shows that between the points a and b the rudder angle of 5 degrees was properly applied by the quartermaster; one notices a corresponding decrease of control action in the automatic apparatus. The judgment of the quartermaster was wrong, however, in bringing the rudder amidships at the point b and still more was it erroneous at the point c when an opposite rudder angle of 5 degrees is introduced for a very short time. It is seen that this error is detected at once by the atuomatic device starting at c a still more intense control action. The error is recognized by the quartermaster at the point e when he puts 10 degrees of rudder, this time correctly. These errors of judgment on the part of the quartermaster are very frequent, especially when he is told to steer as accurately as he can, which probably accounts for a certain nervousness due to an overstrain.

The automatic device on such occasions seemed to have a better nerve than the quartermaster.

FINAL TEST OFF SAN PEDRO.

On September 22nd the adjustments were completed and on September 23rd the final test took place; its duration was about two hours, with slight interruptions due to the changes of course, etc.

From the very beginning, the ship stabilized itself on the course with an initial swing of only two steps on the 10′ repeater compass motor. On the opposite swing, the angle reached only the first step.

Hardly a minute after the automatic steering was started, a telephone inquiry was made from the bridge as to the behavior of the rudder; it was stated that the rudder indicator did not seem to function. An inspection of the mechanical indicator moved from the follow-up gear rack (steering gear compartment) revealed, however, that the rudder was moving at a slow speed over small angles not exceeding, as a rule, 2 degrees or 3 degrees. Under those conditions, the electrical rudder angle indicator, whose first step (on both sides from zero point) was at 5 degrees, naturally could not show anything else but zero on the bridge.

Approximately one minute after starting, the initial transient condition in steering died down and a steady state was reached. During this period, the compass control disc was on zero most of the time; occasionally, and for rather short duration, it was deviated on the first step (10′). From the fact that the control disc was on zero one could not conclude, of course, that the ship was *exactly* on her course. All that could be concluded, under the circumstances, was that the ship was steered with a degree of accuracy of $\pm$ 10′; in a similar manner, on rare occasions when the control disc went to the first step, the conclusion could be formed that the steered course was at that time between zero and 20′ off its set value.

The lack of any more accurate means of ascertaining the angle of departure from the set course than that afforded by the standard Sperry 10′ transmitter-repeater system did not allow an exact

determination of the degree of accuracy of the automatic steering. All that could be concluded under these circumstances was that this accuracy was most of the time within the limits of $\pm$ 1/6 degree and occasionally going up to the limit of $\pm$ 1/3 degree (two steps).

It is interesting to note in this connection that when the compass control disc was on zero (and such was the case most of the time) the compass control was absent and the automatic steering was performed by the gyrometer with the associated accelerational attachment. This was quite in accordance with the theoretical expectations previously set forth; under those conditions the duty of the compass positional control was to correct from time to time the cumulative disturbance as soon as it would reach a value of 1/6 degree; the actual steering however was performed by the dynamical part of the control, viz. angular velocity (gyrometer) and the accelerational control.

Half an hour after the beginning of the tests, the writer, at the invitation of the Commanding Officer, went to the conning tower, where some observations were also made. Very little information could be gathered from the electrical rudder angle indicator for the reasons already explained; only from time time the indication of 5 degrees of rudder angle would appear for a very short time, apparently under the influence of a particularly heavy disturbance. The lack of a more refined rudder angle indicator interfered considerably with these observations.

Approximately one hour after the beginning of the test, the course was changed about 30 degrees left and a somewhat stronger wind appeared from the port beam; the ship began to carry about 3 degrees of rudder and the changes of the rudder angle on the indicator became more frequent. During this second period of the test, the quartermaster was instructed to report any rudder changes which, in his opinion, should be made by observation of the bearings in order to compare his reports with the actual automatic steering. The reports of the quartermaster were lagging approximately by 3–5 seconds behind the actual movement of the rudder operated automatically, which, of course, was approximately the same order of magnitude which was previously ascertained.

One of the most interesting features of the automatic steering, according to the general opinion of those who witnessed the test, was the performance of the hydraulic steering gear. Whereas on manual operation the sudden starting and stopping of the gear is generally accompanied by considerable noise, its automatic performance was almost noiseless; the gear, in fact, was creeping rather at a very slow speed than actually moving. The speed of the control shaft during this period was between 10 and 25 R.P.M. instead of 140-160 R.P.M. of its normal operation by the quartermaster.

As an additional observation communicated to the writer after the tests by the Engineer Officer was the particularly well balanced condition of the load on both main motors as evidenced by the wattmeter indications during the tests. This naturally could be expected in view of the fact that the rudder was used only by very small amounts.

PART III.

CONCLUSIONS—GENERAL PROBLEMS OF DIRECTIONAL STABILIZATION.

The *New Mexico* automatic steering tests, in spite of the limitations previously mentioned, throw considerable light on dynamics of automatic steering and allied problems.

The technical conclusions which can be derived in this manner are very numerous but it is outside of the scope of the paper to present them here in detail.

It is interesting to note briefly that the dynamics of the phenomena involved in the control of angular position of ships (e.g. steered course in case of automatic steering, level condition of the ship in case of antirolling stabilization, etc.) apparently can be formulated in the form of two fundamental principles:

(1) Principle of continuity.

(2) Principle of a dynamically correct phase of the control action.

The first principle is almost self-evident and was emphasized on several occasions in this paper. The electro-hydraulic steering gear of the *New Mexico,* on the one hand, and the controlling action derived from the angular velocity responsive instrument—

gyrometer, on the other hand, were linked together so as to form one single system characterized by a perfectly continuous controlling action on the rudder. The advantages of such a continuous control can be easily understood if one takes into account the continuity in time of the disturbances whose compensation is the primary object of the control. These disturbances never act on the ship abruptly but develop gradually, varying from one instant to another by continuity. The above considerations become particularly clear if one compares the results of the Puget Sound test with later tests. As already mentioned the Puget Sound test was characterized by a considerable predominance of the compass control over the angular velocity control.

As one probable cause of the low efficiency of steering during that test was undoubtedly the discontinuous step by step (10′ of arc) performance of the repeater motor, the control was remaining idle between the steps of the repeater motor and the disturbances were allowed to act during this time without any check on the part of the control. It is necessary to emphasize, however, that the above principle of continuity gives only a *necessary* but by no means *sufficient* condition for an adequate control. In other words, it would be erroneous to conclude that a mere substitution of a more refined repeater motor with 2′ steps (instead of 10′) would improve the results of the Puget Sound test materially because the phase of the control action would not be changed by such refinements in the positional control from the compass.

The second principle of the adequate phase of the control action is less obvious than the principle of continuity.

Let us consider a certain quantity characterizing the magnitude of a disturbance as a function of time. This may be the resultant disturbance torque due to all unknown factors causing the development of the yaw, or of rolling or of any other undesirable angular motion. It is self-evident that if at any moment an equal and opposite moment were applied in the corresponding angular degree of freedom, the disturbance would not affect the system (i.e. the ship) at all. But to accomplish this we need the advice of a kind of supernatural agency capable of informing us a few instants in advance as to the magnitude of the incoming disturbance, its duration, etc.

The absurdity of such a formulation of the problem, however, becomes less apparent if we analyze the properties of continuity of functions. Let us apply an analytical method of reasoning and assume that we are in possession of such a supernatural agency and that the control which it affords is absolutely correct in the above defined sense. For example, in case of automatic steering this control is characterized by the angle of the rudder $\rho(t)$ considered as a function of time. Clearly the only feature of the function $\rho(t)$ which we know about is its continuity. Consider the values of this function at the instant t_o and a short time Δt after this instant. The time interval Δt can be considered small enough in comparison with the time of development of the disturbing motions (yaw, roll) and of the order of magnitude of the time lag in the control transmissions. In the problems relative to the ships it is consequently of the order of magnitude of 1/4, 1/2, or one second of time.

By Taylor's expansion:

$$\rho(t_o + \Delta t) = \rho(t_o) + \Delta t \left(\frac{d\rho}{dt}\right)_{to} + \frac{\Delta t^2}{1.2}\left(\frac{d^2\rho}{dt^2}\right)_{to} + \cdots \quad (2)$$

If the problem considered is the automatic steering and ρ is the rudder angle, the above equation after elementary transformations becomes:

$$\rho(t_o + \Delta t) = \rho(t_o) + \left(\Delta t \frac{d\rho}{d\alpha}\right)\left(\frac{d\alpha}{dt}\right)_{to} + \left(\frac{\Delta t^2}{2}\frac{d\rho}{d\alpha}\right)\left(\frac{d^2\alpha}{dt^2}\right)_{to}$$

$$= \rho(t_o) + a\left(\frac{d\alpha}{dt}\right)_{to} + b\left(\frac{d^2\alpha}{dt^2}\right)_{to} + \cdots \quad (3)$$

$$\text{Where } a = \Delta t \frac{d\rho}{d\alpha};\ b = \frac{\Delta t^2}{2}\frac{d\rho}{d\alpha} \text{ etc.}$$

are certain constant quantities depending on the mechanical connections between the deviation α from the set course and the rudder angle, as well as on the order of magnitude of Δt with which we are concerned.

Equation (3) is of fundamental importance. It shows that the *future conditions of the control (i. e. $\rho(t_o + \Delta t)$ can be foreseen correctly if in addition to the present condition $\rho(t_o)$ of the control*

one knows the present ($t = t_0$) *values of the higher time derivatives of the disturbed motion.*

Theoretically, an infinite number of terms is necessary in equation (3). Practically, one or two are needed in view of a rather rapid convergence of the series if Δt is small enough.

The most interesting feature of equation (3) is that its left member relates to the *future* ($t_0 + \Delta t$), whereas its right member is composed of the terms all relative to the *present* instant of time t_0.

There is no necessity now for a supernatural agency which we have invoked originally; equation (3) shows that the higher time derivatives are capable of giving us the necessary warning as to what is going to happen a few instants ahead of the present time.

The above considerations are of course not new; they relate to general methods of extrapolation of continuous functions.* Their application to the control problems, as far as the writer knows, is new, and the *New Mexico* installation is undoubtedly the first step in this direction. If one reviews the experimental results outlined in Part II of this paper, one cannot fail to notice that the Puget Sound test having coefficient $b = o$ (equation 3) and a very small coefficient a was very unsatisfactory. The Bellingham test was more satisfactory because coefficient a was materially increased. The San Francisco, and especially the San Pedro tests, were still better, owing to the fact that the angular acceleration control (i.e. coefficient b of equation 3) was added. It is true that this accelerational control being *discontinuous* was far from being perfect but nevertheless its addition was responsible for the accuracy of steering obtained during the San Pedro test. There is no doubt that with a *continuous* accelerational control (which is quite possible with modern means), the accuracy of steering could be materially increased as compared with that during the San Pedro test and the rudder angles used could be made still smaller.

In the earlier publication above referred to, it was shown that if the control is made responsive to a linear combination of higher

* As another example of the same nature, one can mention the extrapolation of a continuous curve joining the subsequent distance records (obtained from rangefinders) by the tangent at the last point with the idea of determining the next probable value of the distance.

time derivatives, in view of the linearity of differential equations of motion for very small motions around the equilibrium point, the control acts in such a manner as if *the actual parameters of the system were changed.* For the above reason, one can designate this particular type of control as *parametric.*

The question of the proper phase to be given to the control action is thus reducible to a proper change of the parameters of the system to be stabilized. Thus, for example, the angular velocity control (proportional to the first time derivative) introduces the damping which is almost lacking at the beginning of the yaw of the ship. Likewise, the angular acceleration control (proportional to the second time derivative) is equivalent to a reduction of the effective moment of inertia of the ship, and so on.*

These considerations are quite general and are applicable to any problem of stabilization such as automatic steering of vessels, control of ship stabilization by the Thornycroft method, stabilization of landing decks of the airplane carriers, guns, instruments, etc.

The *New Mexico* installation, from this standpoint, was based on a parametric control utilizing the first $\left(\frac{d\alpha}{dt}\right)$ and the second $\left(\frac{d^2\alpha}{dt^2}\right)$ time derivatives of the yaw.

We now reach a point where the question can be asked: what is the use of having such accuracy in steering when other disturbances due to rolling and pitching cannot be counteracted by the rudder anyhow; and even if there is no rolling or pitching, why have such supersensitive steering? The whole question can be considered in a somewhat different light. The automatic steering undoubtedly can be perfected down to 5′ of accuracy in the steered course. Calculation shows that the same parametric control applied to the antirolling stabilization by the Thornycroft method, for example, very likely will be able to eliminate about 95 per cent of rolling under the conditions of weather, which may be considered rather limiting for a naval action. The ship under those con-

* N. Minorsky loc. cit. page 298.

ditions is a gun platform stabilized in azimuth, say within ± 5′ and in broadside elevation within say ½° — ½° — 1° of the arc.*

The pitching and the forced yawing among waves are thus far omitted from the study.

But let us take a step further and assume that on a pre-stabilized gun platform, which is the ship at this point of our investigation, is installed another high sensitivity parametric control taking care of this residual motion.

It is not proposed here to enter into any technical discussion of this "stabilization of the second order" problem. It is sufficient to mention briefly only some interesting engineering possibilities resulting from such double application of the parametric control. Assume that this second order control is actually applied to azimuth and elevation gears of turrets, guns anti-aircraft batteries, of anything which must maintain its direction in space. If we assume again tentatively that 90 per cent to 95 per cent of the residual disturbances can be eliminated by the parametric control, utilizing again the first and second time derivatives as controlling factors, we reach a conclusion that both the offensive and defensive means of the ship can be stabilized within, perhaps 1′–2′, and this probably will be quite satisfactory from the gunnery standpoint.

There is no doubt that the above order of magnitude can be obtained without much difficulty, owing to the double parametric control (of the first and second orders) utilizing the first and second time derivatives.

A number of conclusions are immediate, the two following being the most important:

(*a*) Owing to the very accurate stabilization (both in azimuth and in elevation) of guns (main battery, anti-aircraft battery and others), it seems perfectly possible to direct the fire from the aircraft. The observing planes, *correcting* the gun fire in this case will be superseded by "the flying conning tower" actually *directing* the gun fire. What kind of connection there will be between the latter and the guns is immaterial. Short waves or infra red radiation can probably be used equally well. The setting points (azi-

* This last mentioned condition depends of course on the amount of the disturbance. The above figure of 95 per cent is taken from a theoretical application of the parametric control of an approximately equal sensitivity as that used during the *New Mexico* trials in connection with the Froude's equation of unresisted rolling among waves.

muth, elevation) of the stabilizing systems can be varied in this manner but the actual stabilization of the axis of the gun fire will be maintained by the parametric controls of the first and second orders.

(*b*) The danger of aerial attacks on the Fleet, so much discussed in recent years, disappears under those conditions entirely. The ships can easily be protected by the smoke screens, not only from the enemy Fleet but also from the above. There is no necessity of "seeing" anything at all from the ships; all "seeing" will be done from the aircraft controlling the gun fire of the Fleet.

It is hoped that this new method of directional control which was designated in the above discussion as parametric will be found of interest to all who are engaged in the study of these problems. The *New Mexico* tests must be considered from this viewpoint as the first step in this general direction.

NAVAL ELECTRICAL POWER IN COMMERCIAL USE—U. S. S. *LEXINGTON* RELIEVES TACOMA POWER SHORTAGE.

By H. L. White, Lt. Comdr., U.S.N., Member.
Engineer Officer U. S. S. "Lexington,"

During the summer and fall of 1929 an unprecedented drought had so depleted the main hydro-electric power reservoir of Tacoma that a power shortage during the approaching freezing weather was imminent. During freezing weather the output at the Nisqually stream-flow plant of the city is greatly reduced; and the reservoir at Lake Cushman or outside power service must then make up the reduction if the demands are to be met. With this reservoir nearly empty and no surplus power available from outside sources, the prospects were none too bright for the continued operation of industries and for an adequate supply of heat for the homes, both of which depended upon electric power. A power shortage thus meant unemployment and suffering from cold.

In view of the situation, the Navy Department ordered the *Lexington* to supply power to the city; and for this power the city agreed to pay to the Disbursing Officer at Navy Yard, Puget Sound, a flat rate of 1/4 cent per K.W.H. for a connected load of 20,000 K. W. plus 1 cent per K.W.H. for the power actually received. As the total power received during the stay from December 17, 1929, to January 16, 1930, was 4,250,960 K.W.H. the cost was $78,509.60 or 01.85 cents per K.W.H. The cost to the ship for fuel and incidental expenses was $18,627.69, leaving for ship's repairs a surplus, disregarding pay of personnel, depreciation, and loss of ship for active service of $59,881.91.

As power had never been supplied from a ship to a city before, there naturally was some difficulty in deciding whether it could be done: and, if so, how it was to be done. The first problem was how to get sufficient depth of water at a dock, not only for

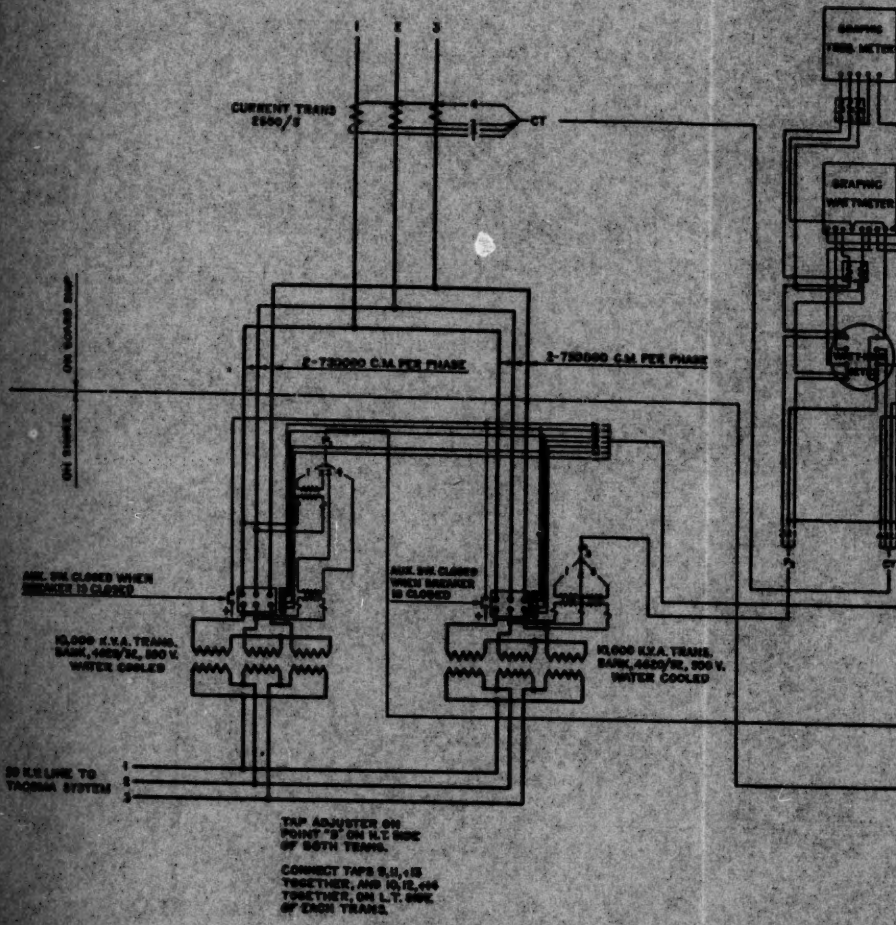
floating the ship, but also for supplying water to the main circulating pumps. These pumps take suction from the bottom of the ship, and each of the four sets has a capacity of more than four million gallons an hour. It was, therefore, apparent that in the vicinity of the pumps a considerable increase in depth beyond the draft of the ship would be necessary. This increase was estimated to be twenty feet; but as no evidence of mud or of recirculation of water was later noted while supplying power, it is now thought that this amount could have been reduced to ten feet, or a total required depth of about forty feet. After consulting the harbor charts and checking the soundings, the depth desired was found at Baker Dock, but was obtained only by breasting out the ship forty feet by means of lighters.

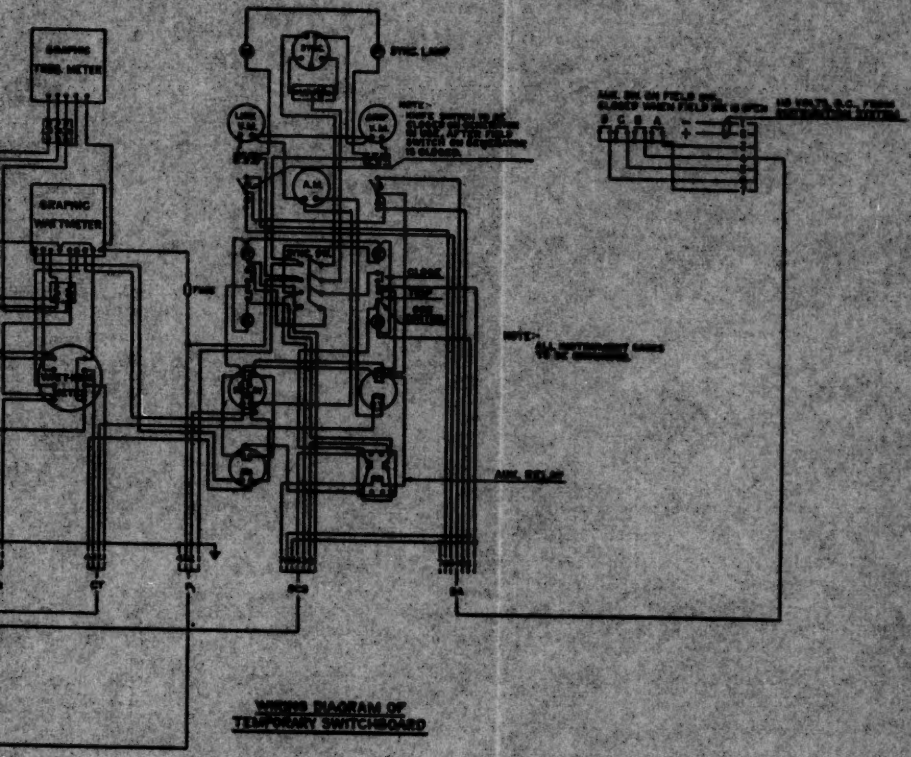
The dock itself had no very substantial means of securing a ship as large as the *Lexington;* and to secure it forty feet out required extra piles ahead and astern and anchors out on the off-dock side. A bow anchor of 34,000 pounds was placed some distance off the starboard bow and two 6,000 pound kedge anchors were carried out and dropped off the starboard quarter.

For the future consideration of problems of this kind, where docks with sufficient depths are not available, the ship could be moored bow and stern and the power cables carried in a bight from the ship's mast to a tower ashore. In cases where the distances were great, towers mounted upon lighters, which would be placed a few hundred feet apart and moored not to turn with the tide, could be used; or, if submersible cables were available, they could be run under water.

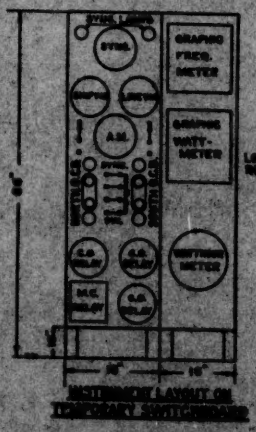
In preventing the ship from swinging with the tide, it could be secured at the bow in the regular way, with one of the anchors, and at the stern by a large buoy, or by planting one of the bow anchors in the proper location, and later picking up and securing its chain to the bitts on either the after starboard or the port gun gallery.

By mooring in the stream, as outlined above, as much power as 340,000 K.V.A., the combined capacity of the *Lexington* and the *Saratoga,* could be made available, not only at Tacoma, but also at Seattle, San Francisco, Los Angeles, Hampton Roads, Philadelphia, New York, and Boston. In the East, where the cities are


1
2
3
CURRENT TRANS 2500/5
CT
2-750000 C.M. PER PHASE
2-750000 C.M. PER PHASE
ON BOARD SHIP
ON SHORE
AUX. SW. CLOSED WHEN BREAKER IS CLOSED
AUX. SW. CLOSED WHEN BREAKER IS CLOSED
10,000 K.V.A. TRANS. BANK, 4400/22, 200 V. WATER COOLED
10,000 K.V.A. TRANS. BANK, 4400/22, 200 V. WATER COOLED
22 K.V. LINE TO TACOMA SYSTEM
TAP ADJUSTER ON POINT "3" ON H.T. SIDE OF BOTH TRANS.
CONNECT TAPS 9,11, & 13 TOGETHER, AND 10,12, & 14 TOGETHER, ON L.T. SIDE OF EACH TRANS.



WIRING DIAGRAM OF TEMPORARY SWITCHBOARD



SWITCHBOARD LAYOUT OF TEMPORARY SWITCHBOARD

U.S.S. LEXINGTON
DIAGRAM OF SHORE CONNECTIONS
AT TACOMA, WASH.

subject to coal strikes, and where the fuel of a great many central stations is coal, the availability of this power might become very important. As oil is pumped to the seaboard in pipe lines from inland as far west as Texas, the supply of oil probably would not be hampered. The ship or ships could, therefore, be sent to Philadelphia or New York, the terminals of the pipe lines, and power as needed could be fed into the transmission system of the city and from there distributed through tie lines at almost any other city in the East.

The six electric drive battleships have a capacity of about 27,000 K.V.A. each or a total of 162,000 K.V.A.—less than one of the large carriers; but due to the low frequency of the power they can not be used commercially except in limited localities where a frequency less than 35 cycles is used, or unless frequency changers are employed. Frequency changers are ordinarily not available.

To take up again the procedure followed at Tacoma, the leads of Number 4 after main motor were disconnected at the motor terminals and connected to vertical bus bars temporarily erected near the motor. By using the connections of a main motor, the power could be furnished from any one of the four main generators and it could be regulated in the control room in the same way as for propulsion. The upper ends of the bus bars were connected to shore leads, which extended 200 feet, through the port after gangway, to circuit breakers on the dock. To prevent the shore cables from dipping in the water, thereby becoming short-circuited, or from breaking due to rise and fall of tide, they were supported on a lighter by a ten-foot timber structure or horse and allowed to hang in a bight. The circuit-breakers, in turn, were connected through transformers to the high tension transmission system of the city. The low-tension or 4620-volt side of the transformers was connected in star and the high-tension or 52500-volt side in delta. The transformers and circuit-breakers were mounted upon two flat cars.

The above equipment had a capacity of 20,000 K.V.A. The three-phase shore leads were 3,000,000 C.M. in area and were made up of twelve 750,000 C.M. conductors. They were insulated with varnished cambric and flame-proof covering for 5000-

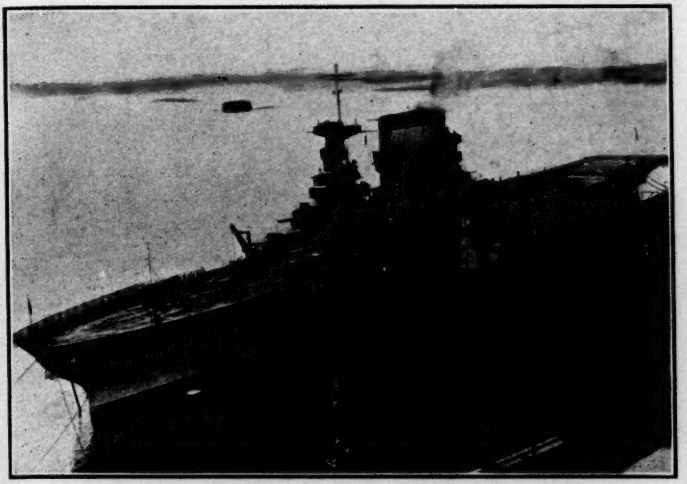
volt service. Each circuit breaker had a capacity of 1200 amperes. The transformers were made up in two banks, each consisting of three single-phase transformers.

The circuit breakers were operated by relays in case of overload, and by relays that were energized by switches on the generator field mechanism, so that when the field was opened in anyway, the circuit-breakers would open also, thus preventing the generator in use from running as a motor. The overload relays were actuated by current transformers installed on the bus-bars in the motor-room and were set at 5000 amperes, which was twice the normal current, in order to take care of surges. The generators were already equipped with ground relays and unbalanced relays, and with these safety features, were considered adequately protected.

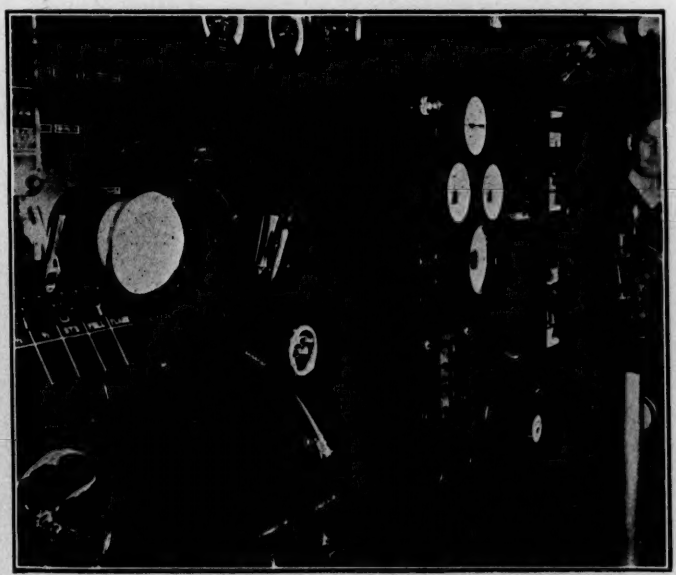
In order to supplement the instruments of the ship, a recording K.W.H. meter, a recording frequency meter, and a Telechron clock were furnished by the city.

For paralleling one of the generators of the ship with the system of Tacoma, a synchroscope, synchronizing lights, a voltmeter on each system, and operating switches for the circuit breakers were installed in the Control Room. The operating switches were placed in a 120-volt D. C. Circuit that energized the relays of the circuit breakers, thereby closing and opening them as desired. As each generator aboard carries its own load and was not designed to run in parallel with other generators, there was some doubt as to whether paralleling was possible. If it were not possible, it would be necessary to divide the city power system so that the *Lexington* could carry an independent load; but, as the surplus power of one part would not be available to the other and as the flexibility of the whole would be reduced, this arrangement was not desirable. Paralleling, therefore, was attempted. After the two lines were synchronized, there was some increased apprehension as to the outcome but when the operating switches were closed, the generator paralleled with so little disturbance of the line that the operator in the city controlling sub-station did not know that the new member of the city system had arrived.

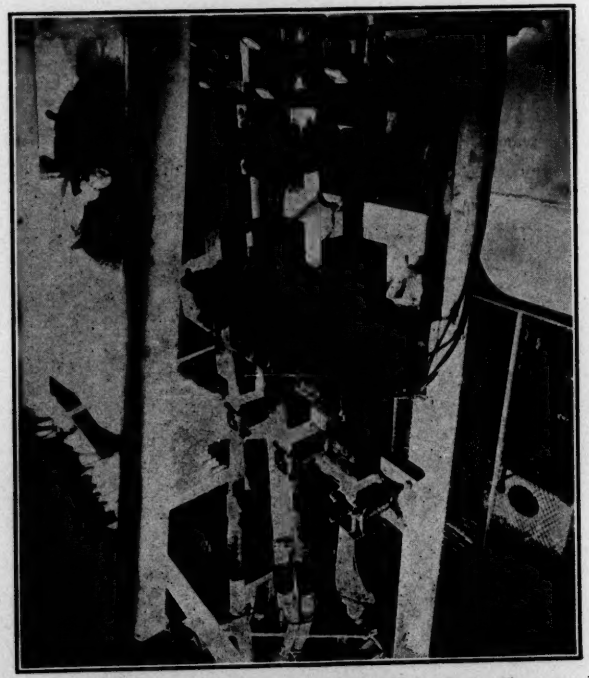
The power was generated at about 4550 volts, 60 cycles, and with a leading power factor to absorb part of the wattless current



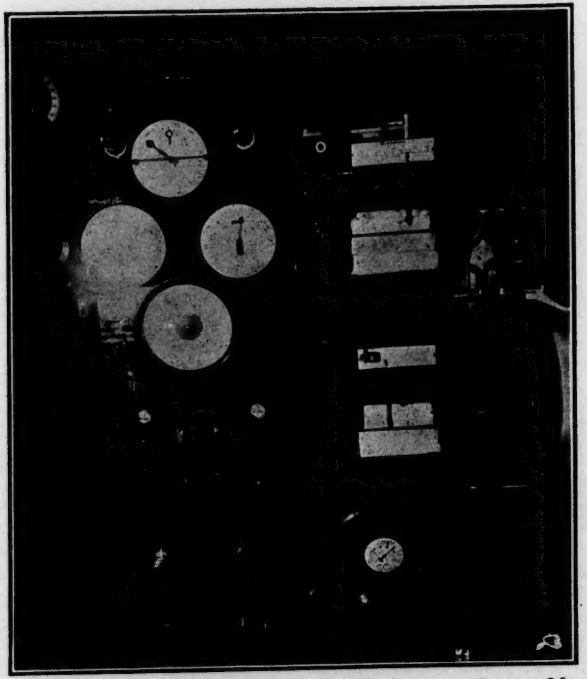
U. S. S. "LEXINGTON" MOORED TO BAKER DOCK, TACOMA, WASHINGTON. POWER WAS DELIVERED TO CITY FROM 17 DECEMBER, 1929 TO 16 JANUARY, 1930.



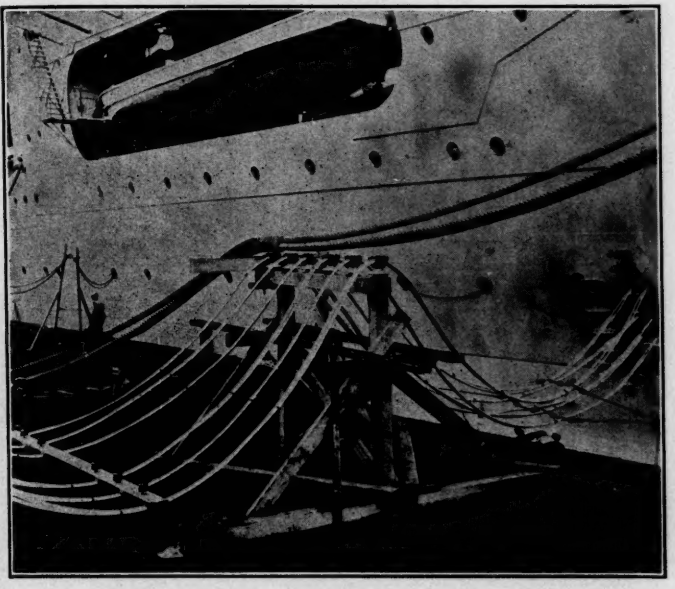
CONTROL ROOM, U. S. S. "LEXINGTON," SHOWING LOCATION OF CITY EQUIPMENT (RIGHT) TACOMA, WASHINGTON. DECEMBER, 1929.



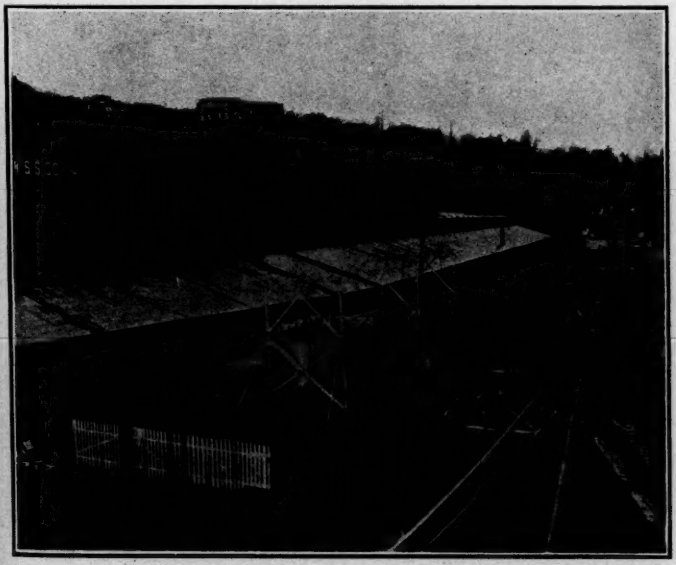
No. 4 Main Motor Room, U. S. S. "Lexington," Showing Electric Leads of No. 4 After Motor (bottom) Connected to Temporary Bus Bars and Shore Leads.



City Equipment for Synchronizing Power System, Power Measurement, and Overload Protection, in Control Room, U. S. S. "Lexington," Tacoma, Washington. December, 1929.



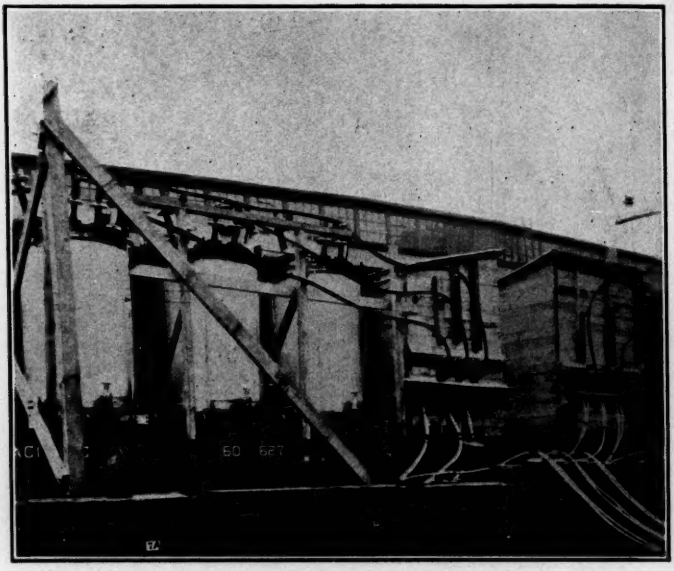
POWER CABLES LEAVING AFTER GANGWAY, U. S. S. "LEXINGTON," AND SUPPORTED BY HORSE ON LIGHTER TO ALLOW SLACK AND KEEP CABLES CLEAR OF WATER DURING RISE AND FALL OF TIDE. TACOMA, WASHINGTON, DECEMBER, 1929.



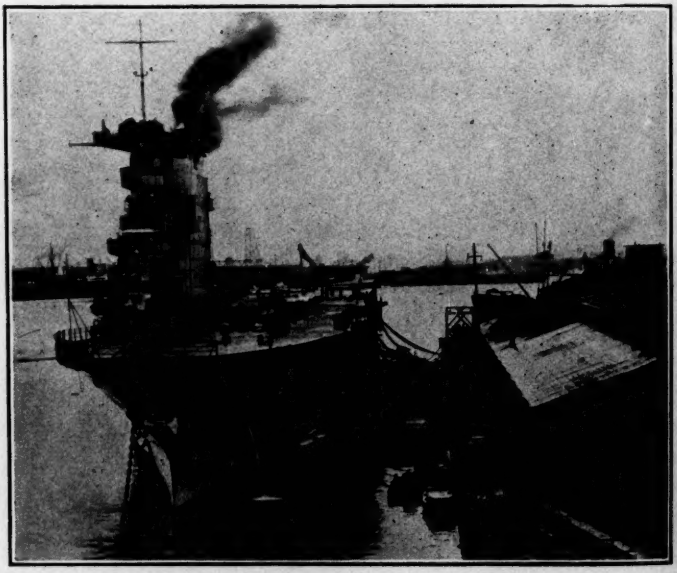
POWER CABLES FROM U. S. S. "LEXINGTON," CIRCUIT BREAKERS AND TRANSFORMERS ON FLAT CARS, AND TRANSMISSION LINE TO POWER SYSTEM OF TACOMA, WASHINGTON. DECEMBER, 1929.

CITY EQUIPMENT FOR SYNCHRONIZING POWER SYSTEM, POWER MEASUREMENT, AND OVERLOAD PROTECTION, IN CONTROL ROOM, U. S. S. "LEXINGTON," TACOMA, WASHINGTON. DECEMBER, 1929.

No. 4 MAIN MOTOR ROOM, U. S. S. "LEXINGTON," SHOWING ELECTRIC LEADS OF No. 4 AFTER MOTOR (BOTTOM) CONNECTED TO TEMPORARY BUS BARS AND SHORE LEADS.



TRANSFORMERS AT END, WITH CIRCUIT BREAKERS BETWEEN, MOUNTED ON FLAT CARS, BAKER DOCK, TACOMA, WASHINGTON. DECEMBER, 1929.



U. S. S. "LEXINGTON," BREASTED OUT BY BARGES TO OBTAIN PROPER DEPTH OF WATER FOR MAIN CIRCULATING PUMPS, BAKER DOCK, TACOMA, WASHINGTON. DECEMBER, 1929.

of the city. During most of the stay the generator of the ship ran on the governor, took the load variations of the city, and controlled the frequency. During the first week the frequency had evidently been a little above sixty cycles as the city electric clocks were found to be ten minutes ahead of Western Union. A Telechron clock, however, was furnished and correct time was then maintained. During the last ten days of the stay the generator ran on the "block" at a constant load of about 20,000 K.W.'s. No casualties, power interruption, or delays occurred and the operations were as satisfactory as would be obtained in a modern central station.

DISCUSSION.

STEAM TURBINE ELECTRIC DRIVE FOR MERCHANT VESSELS.

PAGE 252, MAY, 1929, JOURNAL.

CAPTAIN H. C. DINGER, U. S. NAVY.

This article, while being of service in presenting the advantages of the electric drive, contains various statements and claims which do not seem to be entirely borne out by the actual facts as known to the writer, and which it is believed should be set forth.

The author states that as power increases there is difficulty in obtaining reliable reduction gears. This statement must be questioned. Entirely reliable gears are available for any power yet contemplated. We have entirely reliable gears in U. S. cruisers transmitting 25,000 shaft horsepower, and even larger ones are found in the English Navy. The limit has by no means been reached with 25,000 shaft horsepower per unit. The reduction gears of the *Bremen,* also, appear to have functioned without serious difficulty.

It may be mentioned that there has been, in the last few years, considerable grief with the "single relatively high speed impulse turbine rotor." The design wherein the high pressure part is made small and of higher speed than the low pressure and where the low pressure has divided flow, is to be preferred for high powered marine work. They do not appear to have as much blading trouble.

The statement that water rates on auxiliary steam turbo generators are so low that there is no advantage, as far as fuel costs in using a Diesel generator, seems also more or less surprising when one considers that the fuel required for the steam auxiliary is over two times that required by the Diesel generator.

SPACE.

The writer appears to indicate that by using electric drive less engine room space is required and that machinery is more accessible than with turbo reduction gears. A visit to recently built vessels actually equipped with these installations will usually show decidedly the contrary result. Condensers may be and are placed below turbines in both systems. The turbo reduction gear is lighter, more compact, more accessible, and costs less. The electric drive may have certain other advantages over the turbo reduction gear, but it certainly does not excell in these points. Compare the engine room of one of our electric drive battleships with that on our cruisers built at the same time; also, compare the accessibility.

WEIGHT.

The turbo electric drive is inherently heavier than the turbo reduction gear and requires more space, because it is necessary to have generators, motors, and electric accessories, which outweigh considerably the reduction gears and require more room. All existing electric drive machinery is materially heavier than comparative turbo reduction gear. Electrification of auxiliaries is just as easily practiced in a turbo reduction gear job as in an electric drive job. Here is the field for the Diesel generator that can furnish power independent of boilers.

FUEL ECONOMY.

There has been presented no actual performance at any time wherein the electric drive produces as good fuel economy as a properly designed and built turbo reduction gear, except at very low per cent of full power. There is plenty of accurate comparative data from both merchant vessels and naval vessels, and this data invariably shows the superior economy of the turbo reduction gear installation. It is only necessary to compare the results cited by articles in notes of the May, 1929, number of the JOURNAL "What is the Most Economical Drive for Ships" and "The Propulsion of Ships by Machinery." The best record for the electric drive ship so far in this country appears to be .723 pound of oil per shaft horsepower for all purposes on the *Virginia;* and .715

on the *Viceroy of India* (British). The turbo reduction gear has produced .62 and probably better. Our naval vessels show, at about three-quarters designed power, .84 pound for turbo reduction gear, express boilers without superheat, and about 1.2 for electric drive, B. & W. boilers with some superheat. The advantage of boilers and superheat lies with the electric drive installation, yet the economical performance is considerably below that of the turbo reduction gear.

The electric drive system can be improved in economy, but so can the turbo reduction gear. With proper stage heating, somewhat higher pressure, 400-600 pounds, high superheat, and best boilers available, a turbo reduction gear giving an economy of .52-.55 pound of oil per horsepower for all purposes is entirely feasible.

The statement that turbo reduction gears require three or four times as much lubricating oil as the electric drive is challenged. There is no inherent reason why materially more should be required. The gear teeth do not consume oil. Lubricating oil consumption is primarily a factor controlled by the operating personnel. Oil is lost by leakage, improper handling and by not using the right oil; not by the gears. A differential of 20 per cent might be considered, due to the fact that the electric drive ordinarily has a few less bearings than the turbo reduction gear. I will, however, personally guarantee to run any well designed turbo reduction gear job on the same oil consumption that any comparable electric drive can show.

The electric drive is free from vibration, but so also is the turbo reduction gear. In both cases the noise due to such auxiliaries as forced draft and ventilating blowers drown out the noises made by the main engines. There is a hum to the generator and motor; also a hum to the gear. There is not much choice between them. The electric drive is more flexible. You can operate all shafts with only one generator; i.e., when more than one generator and shaft are present, often not the case on a merchant vessel. The electric drive is slightly more economical at very low percentages of power. This, and the fact that it is more flexible for rapid and variable changes in power, is the real valid reason for using it on battleships, ferry boats, tugs, etc. Our battleships not fitted with

electric drive perform quite well, although their machinery is out-of-date compared to late turbo reduction gear installations. Our new cruisers and some of our auxiliary vessels have very efficient turbo reduction gears.

Our naval electric drive machinery has not been less free from casualties and the need for extensive turbine repairs than the naval turbine reduction gear. Upkeep expenses have not been low; in fact, they have been quite high. In the case of battleships, there has been the idea that the Navy could afford this extra expense on account of certain other operating advantages, but these other advantages do not loom up so great in a commercial vessel.

If it is a matter of dividend paying economy on a high powered merchant vessel, it is believed that turbo reduction gear drive has the edge on the electric drive. There certainly is not much doubt about this if actual performances are properly considered and compared.

It is more or less doubtful whether a little less noise and vibration and a little more convenience in maneuvering will make up for 3 to 10 per cent more fuel consumption, higher cost, and more space and weight. A well built electric drive installation, where fatigue failure of impulse blading is not encountered, will work well and give very good results, but it will not have quite the real operating economy that the best types of turbo reduction gears will give.

It is really surprising how often the erroneous statement that the electric drive apparatus is more economical than the comparable turbo reduction gear installation appears in print, and how widespread this opinion is held. Let us consider the facts that are available.

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THE NORDDEUTSCHER LLOYD LINER "EUROPA."

Some Technical Details of Hull and Propelling Machinery.

The placing in service last month of the new Norddeutscher Lloyd liner *Europa* marks an important milestone in the history of transatlantic shipping. It represents the third, and for the time being the final, step in the progressive post-war policy of her owners, whose purpose has been to provide a weekly express service across the Atlantic. The first step was taken in July last, when the *Bremen** was completed and succeeded on her maiden voyage in recapturing for Germany the coveted Blue Riband held for so long by the Cunard liner *Mauretania*. Less spectacular, but hardly less important for the service in view, was the re-engining of the N.D.L. 32,000-ton *Columbus*,† giving her a sea speed of 23 knots; and with this second step but recently completed, comes the third and final step as represented by the *Europa*. *Bremen, Columbus, Europa!* Surely no greater shipping triumvirate than this has ever graced the service of any individual transatlantic company; and even in an age like the present, when the building of great express liners appears to have become the caprice of some Governments in Europe and in America, considerable time may well elapse ere the express service of the Norddeutscher Lloyd is forced to take second place.

DIFFERENCES BETWEEN THE "BREMEN" AND "EUROPA."

Having been ordered simultaneously, launched within 24 hours of one another, and having had their models tested in the same experiment tank, the *Bremen* and *Europa* in the eyes of the lay observer, and even of the more introspective technical student, may be presumed to be regarded as practically identical in all their major characteristics. In actual fact, however, such is far from being the case, and possibly no two important sister ships have ever been built wherein such widely marked differences in applied naval architecture and marine engineering have been embodied. The *Bremen*, as is now well known, was built and engined at the A. G. Weser yard of the Deutsche Shiff- und Maschinenbau A.G., or Deschimag, her propelling machinery being designed by that great marine engineer, Professor Dr. Gustav Bauer. The *Europa* was built and engined by Messrs. Blohm & Voss, Hamburg, who, as the builders of the *Leviathan* and *Majestic*, have had great experience in the construction of mammoth Atlantic liners. The machinery of the *Europa* was designed by that equally eminent engineer, Dr. Frahm.

Dealing in the first place with the ship as distinct from the machinery, the hull of the *Europa* has been designed along what may be described as more orthodox lines than that of the *Bremen*. In the last-named ship, it will be recalled, the shell-plate landing butts below water were arranged leading forward—a novel and distinctly unorthodox procedure—whereas those of the *Europa* follow standard practice. The Oertz rudder is fitted to the *Bremen*, whereas the rudder of the *Europa* is of the ordinary balanced spade type. While the same attention has apparently been devoted in both ships to the reduction of wind resistance by carefully rounding all superstructures and by avoiding an undue number of exposed ventilator cowls, the "falling raindrop" section of the *Bremen's* funnels has not been adopted in the *Europa*, as her funnels are of very flat but elliptical cross section. The length, breadth and depth of both ships are, there is every reason to believe, identical, but seen in profile there are certain definite contrasts which strike the eye of the keen observer. The bulbous bow of the *Europa* is fuller

* No. 229, Vol. XXXVI, p. 633.
† No. 234, Vol. XXXVII, p. 86.

and more snub-nosed than that of the *Bremen*, and the Hamburg-built ship appears to have a much more pronounced sheer than the *Bremen*, especially forward. This pronounced sheer forward should add to the sea-keeping qualities of the *Europa*, especially at high speed in a head sea. The *Europa* appears to sit lower in the water than her Bremen-built consort; yet, paradoxically, there is a definite impression of greater superstructure height in the *Europa*, especially between the funnels, where the height almost appears to be one deck greater than the *Bremen*. This may be accounted for in part by the fact that the aeroplane and its catapult gear are absent from the last-named ship, although the necessary foundations are already *in situ*. In the *Europa* the cruiser stern appears to be slightly more graceful than that of the *Bremen*. Frahm anti-rolling tanks are common to both ships, as also are the expansion joints on the upper promenade or boat deck, the divided funnel uptakes, and the very large metallic lifeboats in Welin-Maclachlan davits. Shell and deck plating is largely comprised of special quality high-tensile steel. While the block coefficient of the *Bremen* at 0.58 may be taken to be substantially that of the *Europa* also, the builders' naval architect has in each case been given some latitude with regard to the ship's lines, and there are material differences in respect to the latter important characteristics.

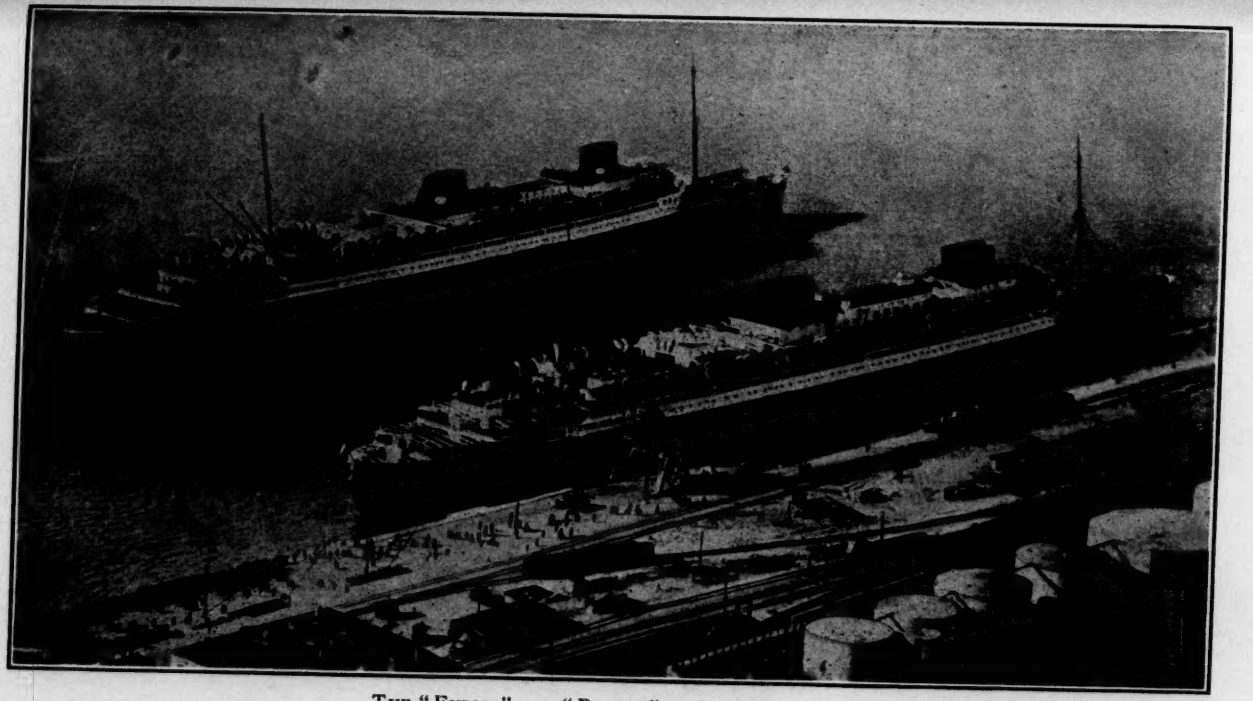
PASSENGER ACCOMMODATION.

The *Europa* provides accommodation for approximately 800 first-class, 500 second-class, 300 tourist third-class, and 600 third-class passengers. The number of tourist third-class passengers, however, can be increased to about 500 by making use of part of the second-class accommodation. These figures are the same as those of the *Bremen*. The crew numbers about 975, so that the ship's total complement is approximately 3175 persons.

The arrangement of the passenger accommodation in the *Europa* is generally similar to that adopted in the earlier vessel, but the schemes of decoration of the public rooms are entirely different. This may be explained by the fact that, whereas the interior decoration of the first and second-class accommodation of the *Bremen* was entrusted to different architects, that of the later ship has been carried out by one architect only—Professor Paul Ludwig Troost, of Munich. The public rooms of the tourist third-class and third-class passengers have been decorated according to the designs of Messrs. Klaus Hoffmann and Friedrich Mezger, of Hamburg. In spite of varieties in the decorative schemes employed throughout the vessel, there is a noticeable general uniformity of style. The paintings, sculptures, tapestries, etc., may well be taken to represent German art abroad. The results achieved throughout have, in our opinion, been most successful, the too ornate note struck in the decorative schemes of so many recent passenger vessels having been avoided. The first-class public rooms include anterooms and reception hall, main dining saloon seating 620 passengers at small tables, two small dining saloons, lounge, art saloon, library and writing room, ballroom, smoking room, children's dining saloon, sun-deck restaurant, and swimming bath. The second-class public rooms consist of a dining saloon, lounge, verandah, ladies' saloon, smoking room and children's playroom. For the tourist third-class passengers there are a dining saloon, lounge and smoking room; and for the third-class passengers, a dining saloon, hall, smoking room and ladies' saloon.

PROPELLING MACHINERY.

That considerable latitude has been given to Dr. Bauer and Dr. Frahm is immediately obvious from an examination of the characteristics of the propelling equipment. Both ships are, of course, propelled by single-reduction



THE "EUROPA" AND "BREMEN" AT BREMERHAVEN.
(The *Europa* is at the Quayside.)

THE BOWS OF THE "EUROPA."

geared steam turbines driving four screws; but whereas all four propellers of the *Bremen* turn outwards, only the two inside or after propellers of the *Europa* turn outwards, the wing or forward propellers turning *inwards*. The theory underlying this most interesting precedent is that the action of the forward inward-turning propellers upon the after screws is more favorable than would be secured by outward turning forward propellers. Two of the *Europa's* working propellers were made in England and two in Germany.

The auxiliary engine-room, situated abaft the after main engine-room, contains, as in the *Bremen*, four 500-Kw. electric generators, each directly connected to a M.A.N. single-acting, four-stroke cycle, airless-injection, six-cylinder Diesel engine rated at 750 B.H.P. These generators supply the current required for heating, cooking, lighting, ventilating, and other auxiliary-power purposes. As in the *Bremen*, comparatively few of the major engine-room auxiliaries are electrically driven. For example, the main circulating pumps—four in number, and each of 8000 tons per hour capacity—are driven by a very powerful compound reciprocating steam engine with balanced cranks. This slow-running, directly-connected engine drives the large-diameter centrifugal impeller at a speed which is not likely to encourage loss of water suction and air locks in the circulating system when the vessel is rolling. The circulating pumps in the *Bremen* are driven by geared steam turbines running at comparatively high speed. The main feed pumps, four in number, with four additional for stand-by purposes, are of Messrs. G. & J. Weir's turbo-driven type, and the four working and four reserve condensate pumps are also turbo-driven. Each main-feed pump is of the rotary type, and is capable of delivering 170 tons of water per hour against a pressure of 470 pounds per square inch when supplied with steam at 315 pounds per square inch. The turbine casing is of cast iron, and the blading and shrouding are of Monel metal, as well as the pump impeller.

As in the *Bremen*, the main propelling machinery is accommodated in two separate engine-rooms, and each of the four turbine aggregates is split up into three turbines—high, intermediate and low-pressure. In common with the prevailing practice in this country, the three-pinion arrangement, or each turbine driving its own independent pinion, was adopted in the *Bremen*, but in the *Europa* an important departure has been made from this principle. The H.P. and I.P. turbines are mounted on a common shaft with a flexible coupling between, and these two turbines transmit their power through a single pinion. On the opposite side of the main gear wheel, the L.P. turbine drives its own separate and diametrically opposed pinion, and the turbines are disposed forward of the main gear wheel. With this arrangement, the very high seating required for the H.P. turbine, which is usually mounted on a level with the top of the large gear wheel, is eliminated, and less vibration should result. Externally the turbines and gears of the *Europa* appear to be less lofty and to take up less space than those in the *Bremen*. There are four main condensers built on the single-flow principle, *i.e.*, the cooling water passes through the condenser once only, and the tubes for all four condensers were supplied by Messrs. Allen Everitt & Sons, Ltd., of Birmingham. Two of the condensers are fitted with Admiralty-mixture tubes and two with cupro-nickel tubes.

THE BOILER INSTALLATION.

Special interest attaches to the boiler installation, which in many important respects differs radically from that of the *Bremen*. In the last-named ship, it will be recalled, there are 20 water-tube boilers in all—11 double-ended and nine single-ended—working on the closed-stokehold system with

a steam pressure of 330 pounds per square inch and superheated to 700 degrees F. The steam characteristics in the *Europa* are the same, but there are 24 water-tube boilers, all of which are of the double-ended type. In this connection it is interesting to learn that all the boilers were built into the ship as soon as the double bottom tank-top plating was finished, and before the frames, shell plating and overhead decks were assembled. This novel constructional procedure was considered to be more advantageous than lowering the boilers down through the uptakes or passing them horizontally through the ship's side; as large water-tube boilers of this type, unlike Scotch cylindrical boilers, cannot be rolled into place without special cradles, and they are, in addition, more susceptible to damage by lifting or handling. The boilers are disposed four abreast and are arranged in four separate stokehold groups. Owing to the narrowing outline of the ship from amidships to forward, the boilers are of progressively smaller size and the number of burners to each boiler front decreases from 6 to 5 to 4 to 3, running forward. This is in contrast with the standard number of seven burners to each boiler front in the *Bremen*.

In addition to the superheaters, which are integral with the tube nests, feed-water economizers are fitted to all boilers in the *Europa*, the economizers being arranged in inverted "V" banks between the sides of adjacent boilers. The economizer tubes are of high-grade chromium alloy steel, and the final feed-water temperature is 150 degrees C., or 302 degrees F. Preheated combustion air is delivered to the furnaces under Howden's system of forced draught; and owing to the presence of economizers, superheater elements and air heaters, induced draught is provided in addition in order to secure the necessary velocity of flow in the combustion gases. Howden's forced-draught fans and the induced-draught fans are directly driven by steam turbines. At the middle length of each boiler and well below the level of the burner fronts, the furnace chamber is internally illuminated by a powerful electric lamp, so that the quantity of smoke and the quality of combustion are at all times under the close observation of the stokehold engineer. In addition, the stokehold is replete with CO_2 and CO recorders, pyrometers, and every other modern device for ensuring the control of combustion. The boiler feed-water regulators were supplied by Messrs. A. G. Mumford, Ltd., of Colchester, and the boiler safety valves are of the Cockburn-MacNicoll improved high-lift patent type as manufactured by Messrs. Cockburns, Ltd., of Cardonald, Glasgow. The same makers also supplied their well-known Cockburn-MacNicoll patent improved emergency shut-off valves, which work in conjunction with Aspinall's patent duplicate turbine emergency governors.

SEA TRIALS AND COMPLETION.

Unfortunately during the builders' trials in the North Sea and when nearing the completion of her 24 hours' full-power test, some trouble was experienced due to the failure of the second-stage impulse blading in one of the high-pressure turbines. It was fortunately possible to shut down the defective turbine before the stripping extended to the reaction blading, and extensive damage was thus avoided. On arrival at Bremerhaven all four high-pressure turbine rotors were removed and transported to Hamburg, and the impulse blading was renewed throughout in a most expeditious manner. The *Europa* subsequently proceeded on a second set of sea trials, which were entirely successful. Her machinery was run for 24 hours at 125,000 S.H.P. without a hitch of any kind, and on the final two hours trial the machinery output was in excess of 135,000 S.H.P. with several

nozzles in reserve. The mean of two runs during the latter trial showed a speed of more than 28 1/2 knots.

The *Europa* left Bremerhaven on her maiden voyage on the 19th March under the command of Captain Nicolaus Johnsen, the commodore of the Norddeutscher Lloyd fleet. The vessel arrived off Southampton the following morning, where, by the courtesy of Messrs. Moxon, Salt & Co., Ltd., the general agents in this country for the Norddeutscher Lloyd, a large number of interested visitors were able to inspect her. After a few hours stay, she proceeded to New York, calling first at Cherbourg.

Prudent and experienced owners, such as the Norddeutscher Lloyd, with the Blue Riband already safely in their keeping for several years to come, are unlikely to encourage the pitting of these two great vessels one against the other. There is, however, the well-known rivalry between Hamburg and Bremen, which has its modified parallel in that between the Clyde and the Tyne in this country, and the friendly competition between the *Lusitania* and the *Mauretania* is still fresh in the minds of most of us. In improving upon the *Bremen's* maiden voyage western crossing by 36 minutes, the *Europa* has already demonstrated her undoubted caliber, especially as the weather by all accounts was less favorable than that enjoyed by the *Bremen*. In addition, the *Europa*, unlike the *Bremen*, followed the winter track, which is some 70 miles longer than the summer route. There is little reason to doubt that neither of these two ships has as yet shown the full extent of her capabilities as to speed; and in due course, and with proper and primary regard for the safety and comfort of the service, we may expect to find first one and then the other improve upon their already remarkable performances. Meantime it is not too much to say that marine engineers the world over will watch with the greatest interest the relative performances of these two epoch-making ships, both of which have added fresh luster to German shipbuilding and marine engineering achievement, and have provided a striking tribute to the courage and enterprise of their owners, the Norddeutscher Lloyd.

* * * *

The contract for the building of the *Europa* was placed in December, 1926, the keel was laid on the 23rd July, 1927, and she was launched on the 15th August, 1928. Her completion was unfortunately delayed by the disastrous fire which broke out on board while the vessel was being fitted out afloat. Notwithstanding the great damage sustained, the ship has taken up her service only about eight months after the *Bremen*.—"The Shipbuilder," April, 1930.

MODERN TURBINE PROPULSION.*

By W. E. Thau.

The purpose of this paper is to discuss modern turbine propulsion in the geared and electric drive forms. By modern turbines are meant the high-speed, high-efficiency turbines arranged for moderately high pressure and high-temperature steam, operating in conjunction with efficient boiler, condensing, and feed-heating plants and, as far as economically practicable, electrically-driven auxiliaries. These two modern forms are thus distinguished from the older and less efficient forms of direct-connected and low-speed single-reduction geared equipment.

* Abstract of a paper read in New York, at the thirty-seventh general meeting of the American Society of Naval Architects and Marine Engineers.

The first turbines applied to ship propulsion were direct-connected. There is even a recent example of this method.† The direct-connected turbine at best is a compromise which is not justified to-day. All factors operate against its performance: the speed is too low for good efficiency; the blade clearances are relatively excessive, resulting in wasteful leakage; moreover, since the direct-connected turbine is of necessity a large unit, the danger of distortion with consequent blade-rub is great, the weight and size are excessive, and such turbines require high skill in handling and maneuvering. In the effort even to approach high speed, at which the turbine is most efficient, there is the constant temptation to sacrifice the propeller efficiency. The direct-connected turbine has served its purpose in the development of the art, and will probably not be used again in a new installation.

To overcome the objections of the low-speed direct-connected turbine it was necessary to introduce speed-reducing means. Two general methods are now available, namely, the reduction gear and the electric drive. Incidentally, both are considerably more than mere speed-reducing means because of their flexibility in emergency reserve power, &c.

GEARED TURBINE DRIVE.

The first geared turbines employed a single-reduction gear. These, however, due to low propeller speeds, were recognized as not representing the most economical arrangement, because the turbine speed was still below its most efficient value. Ninety revolutions per minute of the propeller necessitated an inefficient turbine speed with single reduction, as the highest practicable reduction ratio has been considered to be approximately 20:1. This led to double-reduction gears and turbine speeds of about 3600 R.P.M. Unfortunately, some of the first double-reduction installations gave trouble, and their adherents in Europe reverted to the single-reduction arrangement. American engineers, however, analyzed the situation differently and overcame the troubles. Today the modern American practice favors double reduction in the case of low-speed propellers.

Naturally, in cases where the propeller speed is high or where the power per screw is high in combination with reasonably high propeller speed, single reduction is entirely suitable and advisable. The chief requisite is to operate the turbine at its highest practicable speed, and this does not always require double-reduction gears. The turbine as an efficient power-producing means is inherently a high-speed machine.

With due recognition of stresses and factors of safety when dealing with known materials, a high-speed turbine is inherently a better proposition, from a cost, operating and maintenance standpoint, than a low-speed turbine for the following reasons:—(1) It is smaller and thus stiffer and more rigid; (2) it has relatively greater blade clearance per inch diameter; (3) it has a relatively higher first critical speed; (4) it has fewer blades; (5) it has less likelihood of defective workmanship; (6) it is less susceptible to distortion; (7) it is less apt to become misaligned; (8) in the multiple-cylinder arrangement fewer cylinders are needed in the higher speed arrangement; (9) smaller and more rugged construction is conducive to greater flexibility in maneuvering, in that the turbine is able to successfully withstand greater abuse such as is caused by sudden starts and stops.

In addition to its better economy, the high-speed turbine has decided advantages in cost, weight and space, all of which have their economic relations to operation of a ship. The advantages of high-speed turbines apply

† The French liner *Ile-de-France*, built in 1927.—EDITOR M. E.

equally to geared and electric drives. In the case of geared drive the turbine usually is less influenced by other items of the transmission, and hence is less restricted.

There has been considerable research in connection with turbine blade materials, blade forms, blade fastenings, blade lashing, blade breakages, blade services, blade stresses, &c., during the past ten years. The results of this research are astounding. Where blade-tip speeds of 500, 600 or 700 feet per second were considered as limits a few years ago, we not only find ourselves using blade speed to-day almost twice the former limit, but at stresses and factors of safety of the same relative value. The old low-speed turbine has been revolutionized, thanks to the pressure of economy. Strange as it may seem, the high-speed, drum-supported reaction blades are not the cause of any concern to specialized turbine manufacturers, nor to those who operate them.

Proper elasticity is desired in any form of drive. Vibrations are to be eliminated and guarded against to the fullest extent. Having designed, built and balanced a geared turbine equipment so as to effect vibrationless operation in itself, there is only one other source of probable disturbance, and that finds itself in the unbalance of the propeller or the performance of the propeller with respect to the action of the water and the resultant combined relation of the two on the stern of the ship. Torsional vibration is the only type which the geared arrangement will transmit to the ship that the direct-drive turbine and the turbine electric drive will not. The two latter drives will dampen this type of vibration, whereas the former will not. Torsional vibration mentioned in gear drives has never proven to be serious. Vibrations due to unbalance of propeller or to other causes connected with the relative action of the propeller, such as the blades operating in the proximity of the hull, &c., will result in rough operation with any type of drive. This obviously is the problem with which the naval architect is concerned.

Properly-controlled elasticity in the drive is, therefore, a much-desired characteristic. In the geared drive this elasticity can be incorporated in the form of a flexible drive shaft between the turbine and the gear. Torsional elasticity is also desirable from the standpoint of shocks resulting from the striking of submerged objects by the propeller. There are cases on record where anchor chains have become entangled with the propellers and forces developed which were sufficient to move the gears off their foundations and partly shear the line shaft coupling bolts, without resulting in injury to the gear teeth or the turbine. Also, propeller blades have been broken off by striking submerged objects, and the impact was sufficient to partly shear the coupling bolts with no damage to gear teeth or turbine.

A further important advantage of the flexibly-supported pinion frame in conjunction with the flexible drive shaft is that it permits the pinion to take a position with respect to the gear which is nearly proportional to its torsional deflection, and hence assure practically uniform pressure along the entire face width. Such distribution of load is impossible with the fixed bearing gear except at one load and one condition of alignment. What happens in the case of the flexibly-supported pinion is that the pinion is free to rotate a small amount about an axis centered in the web of the "I" beam and thus avoid concentration of the load on one end of the pinion. Greater designed tooth pressures are possible with this arrangement than with the fixed bearing gear, not only with equal but better service of the installation. Consequently the result is not only reduction in size and weight of the gear, but the provision of means to correct for reasonable misalignment, two decidedly beneficial characteristics.

Broadly speaking, the following are the most economical arrangements of geared turbines for vessels of the usual requirements.

Total S.H.P.	No. of Screws.	Propeller R.P.M.	Gear Arrangement	No. of Pinions.	Turbine Arrangement.
8,000	1	80	Double reduction	2	Cross compound.
40,000	2	125	,, ,,	2	,, ,,
100,000	4	150	,, ,,	2	,, ,,
160,000	4	200	Single reduction	3	Triple expansion.
200,000	4	225	,, ,,	4	Double cross compound or quadruple expansion.

In cases where it is desirable to have good economy over a wide range of power and speed, the triple-flow arrangement is the most advantageous in that its flexibility affords a practically flat water rate performance from 100 to 10 per cent rated S.H.P. and 100 to 50 per cent ship speed. In a specific case of 20,000 S.H.P. per screw, the maximum and minimum performance over this range differed less than 3 per cent. In most cases, however, the cross-compound arrangement gives reasonably flat performance over the range at which any ordinary commercial ship is required to operate.

In case of casualty to one of the turbine units or to one of the high-speed gear reductions it is possible to develop approximately 60 per cent power with the remaining element of a cross-compound arrangement, and approximately 75 per cent power with the two remaining elements of a triple-expansion arrangement. In this respect the geared turbine arrangement possesses attractive flexibility in reserve power in case of casualty to any part except the main gear.

The turbine in the case of the electric drive differs from that in the gear drive in the following respects:—

(*a*) The operation is in only one direction, and hence the turbine has no astern element.

(*b*) The complete expansion of the steam is effected in one cylinder. This condition in the large powers necessitates lower speed turbines than in the case of multiple-cylinder geared units. This is further influenced in some cases by the practice of operating below the critical speed. In the case of the reaction turbine generator set the generator critical is lower than that of the turbine, and from this standpoint the generator critical speed is usually the determining factor. By operating through the critical speed some saving in cost, weight, space and economy would result. Mechanically, the turbine generator set is arranged with three or four bearings, the latter arrangement being used in the larger sizes.

TURBO-ELECTRIC DRIVE.

Thus far, no attempt has been made to use other than a complete expansion turbine in the case of electric drive, although both the tandem compound and the cross compound have been considered, the former with a single generator and the latter with the generator arranged in two parallel-

operated units. Parallel-operated complete-expansion units have also been given consideration in certain cases of single-screw ships where reserve power in event of casualty was regarded as mandatory. This arrangement, however, adds considerably to the expense and, besides, complicates the operation and arrangement. It is not recommended.

Substituting a cross-compound arrangement, in which each unit is of half-capacity, for a complete-expansion arrangement of full capacity would permit higher turbine and generator speeds, with consequent improvement in economy; however, it entails complication in operation and additional space, weight and cost, which more than counteract the gain in economy. The tandem-compound arrangement merely provides reserve power in the event of casualty to a turbine, and has little, if anything, to commend it.

All things being considered, it appears as though the best turbine arrangement for electric-drive installations is to use complete-expansion turbines, one generator set for each propeller motor in the case of twin-screw ships. In the case of quadruple-screw ships, two or four generator sets are used, depending largely on the power and flexibility desired. When driving inboard and outboard screws with one generator set, it is either advisable to design the inboard and outboard screws differently so that they will divide the power equally, or to incorporate additional power in the inboard or all motors. One or the other of these arrangements is necessary, because the inboard screws, when of the same design as the outboard screws and operated at the same speed, will absorb more power than the outboard screws. The amount of difference in the power absorbed between the inboard and outboard screws depends, obviously, on the relative wake action. Cases are on record in which the inboard screws absorbed approximately 15 per cent more power than the outboard screws when operated at the same revolutions per minute. Since the synchronous motor is suitable for at least all ordinary ships, that form alone has been considered in this discussion. The induction motor drive is heavier, more expensive and slightly less efficient in operation; however, experience has shown that is not required in any except unusual ships.

During the past year the marine technical Press has been replete with articles dealing with the merits and demerits of the two systems under discussion. In discussing a subject of this character it must be clearly recognized that both types have certain inherent good characteristics which might be overlooked, or not properly evaluated. The electric drive advocates stress such factors as reliability, maintenance, ease of control, maneuverability, astern power, vibration, &c. The geared turbine advocates emphasize some of the same factors, and include such items as first cost, operating cost, less complication, weight, space, &c. The major factors under discussion are briefly analyzed below.

ELECTRIC AND GEARED DRIVE.

(*a*) *Dependability or Reliability.*—This item is probably the most important. When all factors are considered, I believe there is little, if any, difference between properly-designed and constructed modern drives of either type. In making a comparison of the reliability of two drives we must be careful to adhere to modern types and not go back to the early installations which in reality are steps in the development, and this applies in both cases. The proof lies in the fact that there are plenty of successful installations of both types in service.

(*b*) *Maintenance.*—This item in a sense is essentially a corollary of dependability. Here an unbiased search, considering all cases on their

merits and in proper proportion, will show about an equal standing, particularly in the more modern installations. An item of improper design, material, construction or operation should be duly recognized as such.

(*c*) *Operation.*—Either type properly operated will give satisfaction. It is a case of knowing the installation and its limitations.

(*d*) *First Cost.*—The first cost, when based on modern efficient machinery, is for the ordinary type of ship in all cases in favor of the modern, efficient geared turbine. Strange as it may seem, the costs of the older and less efficient arrangements of geared turbine drives using low-speed turbines are more nearly in line with those of the turbine electric.

(*e*) *Efficiency.*—Comparing turbine with turbine in the two types of drive, the performance is at a stand-off, *i.e.*, for the same steam and vacuum conditions the Rankine cycle efficiencies on a turbine B.H.P. basis, neglecting the astern element, are about the same when an honest effort is made to get the best in each case. However, when including the astern element, transmission efficiency and auxiliary power requirements, such as excitation, ventilation, lubrication, &c., all as considered in the overall fuel rate per propeller S.H.P. hour, the modern efficient geared turbine is the more efficient. The percentage difference in fuel rates is less in the case of the larger drives.

(*f*) *Operating Cost.*—This item combines the investment charges on the first cost and the fuel cost, and is in favor of the geared turbine. The exact amount varies and is influenced by many factors. The difference is appreciable in some cases and relatively inconsequential in others. A full and detailed comparison of this item would be very complicated and likely indefinite when made in any except a specific case. The investment cost differential is smaller in the case of the lower powers and greater in the case of the higher powers. On the other hand, the fuel cost difference is a greater percentage in the lower powers than it is in the higher powers.

(*g*) *Astern Power.*—Much has been said concerning the relative performance of the two drives in astern power. Geared turbine arrangements will supply anywhere from 25 to 70 per cent H.P. astern, depending upon the design and character of the astern elements. Electric drive, using a single winding motor, will supply 100 per cent astern H.P. at the motor shaft. Electric drive, using two separate pole combinations in the motor windings, will supply from, say, one-third to full H.P. astern, depending upon the control arrangements. There are ships of all mentioned astern characteristics for both types in service, and their astern power seems to be ample. It is true that ordinary ships are not built to run astern at full power, else the propeller would be symmetrical for ahead and astern operation, or bow and stern propellers would be installed such as is the case with double-ended ferryboats. Ample torque is the necessary characteristic for quickly stopping and reversing a ship, and this is available in the required amount in both types. Service demonstrates this. It is a fact that the modern efficient geared turbine will turn to its full astern R.P.M. in the same or less time than the modern efficient electric drive. Since full H.P. can be developed in either direction by the electric drive, higher astern revolutions per minute can be obtained. With the proper control arrangement the modern efficient geared turbine will handle as easily as the electric drive.

(*h*) *Complication.*—To those who understand electrical apparatus there is nothing to the argument of "complication." To the electrical man, the mechanical drive looks complicated. It is true that the turbine power goes through a number of carriers, in the electric drive, before reaching the propeller shaft, but in the final analysis they are all simple enough, as well as reliable, when understood and cared for.

FUEL RATE ANALYSES FOR GEARED TURBINE AND TURBO-ELECTRIC DRIVES.

FUEL RATE CALCULATIONS FOR 5,000 S.H.P.—SINGLE SCREW.

	Geared Turbines.			Turbine-Electric.	
	A.	B.	C.	D.	E.
Condensate temperature .... ° F.	88·0	88·0	88·0	88·0	88·0
Rise in air-ejector cooler .... ,,	10·6	10·6	11·1	10·0	10·3
Rise in feed tank .... ,,	24·7	24·9	25·0	24·8	24·7
Rise in low-pressure heater .... ,,	81·7	81·5	80·0	82·2	82·0
Rise in high-pressure heater .... ,,	95·0	95·0	95·0	95·0	95·0
Final feed-water temperature .... ,,	300·0	300·0	300·0	300·0	300·0
Steam to air ejectors .... Lb./hr.	400	400	400	400	400
Steam to main condenser .... ,,	36,000	37,500	36,000	37,000	39,000
Steam to auxiliary generator .... ,,	1,700	—	*	3,130	—
Steam to main feed pump .... ,,	1,750	1,750	1,750	1,750	1,750
Steam to miscellaneous heating, &c. .... ,,	2,000	2,000	2,000	2,000	2,000
Steam bled to low-pressure heater .... ,,	1,630	1,610	1,450	1,900	1,775
Steam bled to high-pressure heater .... ,,	4,800	4,800	4,600	5,100	4,950
Steam from boilers .... ,,	48,280	48,060	46,200	51,280	49,875
Power developed by main units .... S.H.P.	5,020	5,070	4,995	5,035	5,000
Power developed by auxiliary generator .... kW.	100	100	100	184	184
Boiler efficiency .... Per cent.	85	85	85	85	85
Heat value of fuel .... B.Th.U./lb.	19,000	19,000	19,000	19,000	19,000
Heat rate—steam only ....B.Th.U./S.H.P./hr.	10,425	10,270	10,025	11,040	10,820
Fuel rate—all purposes .... Lb./S.H.P./hr.	0·646	0·637	0·633	0·685	0·670

FUEL RATE CALCULATIONS FOR 30,000 S.H.P.—TWIN SCREW.

Condensate temperature .... °F.	88·0	88·0	88·0	88·0	88·0
Rise in air-ejector cooler .... ,,	5·9	6·0	6·3	5·7	5·7
Rise in feed tank .... ,,	23·6	23·7	23·7	23·6	23·8
Rise in low-pressure heater .... ,,	87·5	87·3	87·0	87·7	87·5
Rise in high-pressure heater .... ,,	95·0	95·0	95·0	95·0	95·0
Final feed-water temperature .... ,,	300·0	300·0	300·0	300·0	300·0
Steam to air ejectors .... Lb./hr.	1,300	1,300	1,300	1,300	1,300
Steam to main condenser .... ,,	202,000	218,000	204,000	205,000	225,000
Steam to auxiliary generator .... ,,	19,400	—	†	23,600	—
Steam to main feed pump .... ,,	5,500	5,500	5,500	5,500	5,500
Steam to miscellaneous heating, &c. .... ,,	5,000	5,000	5,000	5,000	5,000
Steam bled to low-pressure heater.... .... ,,	16,400	16,000	14,500	17,200	16,750
Steam bled to high-pressure heater .... ,,	27,700	27,200	25,500	28,500	28,100
Steam from boilers .... ,,	277,300	273,000	255,800	286,100	281,650
Power developed by main units .... S.H.P.	29,650	29,610	29,680	29,550	29,400
Power developed by auxiliary generator .... kW.	1,500	1,500	1,500	1,830	1,830
Boiler efficiency .... Per cent.	85	85	85	85	85
Heat value of fuel .... B.Th.U./lb.	19,000	19,000	19,000	19,000	19,000
Heat rate—steam only ....B.Th.U./S.H.P./hr.	10,150	10,000	9,350	10,510	10,390
Fuel rate—all purposes .... Lb./S.H.P./hr.	0·629	0·619	0·610	0·652	0·644

* 62 lb. fuel oil per hour to Diesel engines. † 920 lb fuel oil per hour to Diesel engines. A and D = Auxiliary current supply from condensing turbine-driven generators, B = Auxiliary current supply from generators driven by the main unit; C = Auxiliary current supply from Diesel-driven generators; E = Auxiliary current supply through transformers from main units.

(*i*) *Weight and Space.*—The weight of a modern, efficient geared turbine drive (not including boilers, condensers and auxiliaries) is about 50 to 60 per cent that of the electric drive. Also, the net floor space is usually less for the geared turbine. However, the items of weight and space are not always important. The machinery arrangement is in many cases favorable to the electric drive, both from the hull standpoint and also from the piping standpoint. A definite evaluation of these items can only be determined in connection with a specific case.

(*j*) Racing of propellers when leaving the water is purely a matter of turbine governor and can be avoided easily in either case by providing the turbine with a speed governor. Such a governor is essential with the electric drive, and is also supplied with certain makes of geared turbine drive.

INTANGIBLE ADVANTAGES.

These favor the electric drive on the following counts:—

1. Torsional vibrations, when and as they may be set up in the propeller and line shafting, are not transmitted to the ship structure.

2. In the case of casualty to a prime mover the shift to the emergency set-up can be made in a few minutes, whereas in the geared drive considerably more time would be required. This is relatively important, particularly in comparison with a geared installation in which the emergency piping and valves are not installed.

3. Unidirectional turbines with consequent simplification and elimination of the astern element.

4. Shutting down of one prime mover unit for inspection or repair to its condenser. If divided water boxes are used, tubes may be plugged without shutting down the turbine, but no extensive repairs can be made while the turbine is operating.

5. The balancing of turbine rotors after installation, and like adjustments, are more conveniently made.

For the fuel rate analyses two arrangements are considered: first, a 5000 S.H.P. single-screw vessel, and, second, a 30,000 S.H.P. twin-screw vessel. The fuel rates are computed for turbine-electric and geared turbine propulsive machinery, and in both cases the analyses show the differences to be expected using dual drive (turbine-electric) or attached generator (geared turbine), and using Diesel-driven auxiliary sets instead of turbine-driven auxiliary sets.

The following steam conditions and feed temperature used in the analyses are regarded as very conservative, and it is recognized that higher pressures and temperatures are entirely feasible in both types of drive:—

400 pounds per square inch (gauge) boiler pressure.
375 pounds per square inch (gauge) throttle pressure.
250 degrees F. superheat at the turbine throttle.
693 degrees F. total temperature at the throttle.
28 1/2 inches vacuum at the turbine exhaust.
300 degrees F. feed water temperature, employing two-stage feed heating.
85 per cent boiler efficiency.
Calorific value of fuel oil, 19,000 B.T.U. per pound.

It is assumed that the drop in pressure between the boiler and turbine of 25 pounds per square inch will just balance the radiation losses and develop sufficient heat to maintain a constant superheat.

In any analyses of economy it is entirely correct to consider the propulsive plant together with its direct auxiliaries and accessories as representing one phase, and the ship's plant such as lighting, heating, cooking, ventilation, &c., as another phase. The ship's plant will have no appreciable influence on

the analyses of the propulsive plant except the cost rate at which auxiliary power is supplied, and this can be duly allowed. Therefore, since the ship's plant load varies so widely for different types of ships and different trade routes, an average condition has been estimated and introduced into the analyses in order to get a better idea of the actual total. The fuel rate tables are self-explanatory; however, it is of interest to note the relatively small saving effected by the use of the dual drive and attached generator. This is to be expected in the case of the 30,000-S.H.P. ship. It is also of interest to note that, considering first cost and fuel cost, Diesel-driven auxiliary sets are not justified in the cases cited.—"The Marine Engineer and Motorship Builder," March, 1930.

MARINE ENGINEERING.

By Rear Admiral R. S. Griffin,* U.S.N.

The greatest progress in marine engineering during the two decades immediately following the foundation of the A.S.M.E. was that made in the development of the triple-expansion as a substitute for the compound engine, which was in general use until about 1885. Its adoption was facilitated and somewhat hastened by the progress that had been made in the manufacture of steel; and curiously enough, that progress was due in no small measure to congressional legislation affecting the construction of naval vessels. The Act of August 5, 1882, contained a requirement that the two vessels authorized should be constructed of steel of domestic manufacture having certain physical characteristics. The Act of 1883, which provided for two more ships, contained a similar clause, but as the size and the power of the ships increased, it became increasingly difficult to obtain the large hollow forgings required for our cruisers, a situation which was recognized in the Act of 1886, which authorized the Secretary of the Navy to purchase such forgings abroad provided he should find it impossible to obtain them in this country in time for the completion of the cruiser *Baltimore*, which was authorized in that Act. Then followed the Act of 1887, which contained a provision that forgings for guns and armor should also be manufactured in the United States; and it was these two Acts that were instrumental in bringing about the heavy forging plants of the Bethlehem Iron Company and the Midvale Steel Company, which made us independent of foreign sources of supply for forgings.

The four vessels authorized by the Acts of 1882 and 1883, the *Atlanta*, *Boston*, *Chicago*, and *Dolphin*, were built by John Roach & Son, the hulls and boilers at their Chester works, and the engines at their Morgan Iron Works, New York. They constituted the beginning of what at that time was popularly termed "the new Navy." The engines of the *Atlanta* and *Boston* were of about 3500 horsepower, and were horizontal, compound, back-acting, with all pumps worked from the side or tail rods. The steam pressure was 90 pounds. The *Chicago* had twin overhead-beam, compound engines of 5000 horsepower and externally fired cylindrical boilers, the entire equipment being similar to that of the Morgan Line steamship *Louisiana*.

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It was the design of Mr. Herman Winter, a member of the Naval Advisory Board, under whose general direction and supervision the four vessels were constructed. The dispatch boat *Dolphin* had a vertical compound engine of 2300 horsepower, built in general conformity with the merchant-ship engine of that time except that there was a horizontal combined air and circulating pump, and that no pumps were worked from the engine.

EARLY STAGES OF NAVAL DEVELOPMENT.

As the upbuilding of the Navy had its reflex in the construction of machinery for the merchant marine, it seems appropriate to give space to the early stages of naval development. Following the Roach ships, the contracts that had been entered into up to 1886 had been with the William Cramp & Sons Ship and Engine Building Company for the construction of the cruiser *Newark* and the gunboat *Yorktown*, and with the Union Iron Works for the cruiser *Charleston*. The *Newark* had horizontal twin-screw triple-expansion engines of 8000 horsepower, and the *Yorktown* similar engines of 3400 horsepower. The *Charleston*, whose plans were purchased in England, was a duplicate of the Japanese cruiser *Naniwa-Kan*, which was built in England, and whose slightly inclined twin-compound engines were said to have developed more than 7500 horsepower. As this seemed impossible of attainment under the conditions obtaining on the trials of our ships, and as the builders of the *Naniwa* declined to furnish a set of indicator diagrams taken on her trial, the contract for the *Charleston* was made on the basis of 7000 horsepower. The contractors were not able to reach that figure, and it was only after making a number of costly alterations in the original plans that results satisfactory to the Navy Department were obtained.

Another of the early vessels whose plans were purchased abroad was the cruiser *Baltimore*, whose engines were designed by the well-known firm of Humphrys & Tennant, London. As in the case of the *Charleston*, the Secretary of the Navy thought it advisable to purchase plans abroad with a view to acquainting our designers with what was supposed to be the latest and best of British designs. The engines of the *Baltimore* were twin-screw, horizontal triple-expansion, of 9000 horsepower, working with a boiler pressure of 135 pounds. Her first trial fell short of what was expected, but after some modifications the contract requirements were easily met.

The next cruisers were the *Philadelphia* and the *San Francisco*, though the armored vessels *Maine* and *Texas* had been authorized a year earlier. The plans of the *Texas* were purchased in England, but the machinery plans of the *Maine* were the design of the Bureau of Steam Engineering of the Navy Department. The machinery of the *Philadelphia* was designed by the Cramp Company, and that of the *San Francisco* under the direction of Chief Engineer George W. Melville, who had as collaborators Passed-Assistant Engineers John C. Kafer and Asa M. Mattice. The Secretary of the Navy laid down a condition that the size of the engines of the *Baltimore* should not be departed from, so both groups of designers retained the diameter of her cylinders, but the *Philadelphia* increased the boiler pressure to 160 pounds, used piston instead of flat slide valves, and Marshall valve gear instead of Stephenson link motion. The *San Francisco* provided interchangeable crankshafts, separate combustion chambers for each furnace, and ashpit instead of the closed-stokehold system of forced draft. Independent air pumps were used in each case instead of pumps operated from the main engine. On trial, the *Philadelphia* developed 8815 I.H.P., and the *San Francisco* 8913. Their completion marked the passing of horizontal engines in our naval vessels, as also the practice of purchasing plans abroad. Before

the completion of these two ships, two gunboats, the *Bennington* and the *Concord*, were built under contract with N. F. Palmer & Co., the hulls at the Roach shipyard, Chester, and the engines at the Quintard Iron Works, New York. They were duplicates of the *Yorktown*.

The effect of this building for the Navy was to give an impetus to the building of triple-expansion engines for the merchant marine. The advance in the manufacture of steel made possible much higher pressures than had hitherto been carried, with economy of fuel and reduction in maintenance cost far below the figures which prevailed with compound engines. The transition resulted in getting a much smoother-running engine, and this with piston speed considerably higher than had been the practice up to that time. The high piston speed was specially noticeable in naval engines, where the urge for reduction in weight per unit of power made it necessary to operate the engines at a greatly increased rate of revolution, and to use high-grade forgings and steel castings wherever their use would contribute to that end. The development that contributed most to reduction in weight was forced draft, applied in most naval vessels on the closed-stokehold system, though there were some with ashpit installations. Up to this time, a steam jet in the smokepipe was the recognized method of increasing the rate of combustion.

Developments that contributed to economy were the use of steam-jacketed cylinders, which followed the experiments on steam jacketing conducted by Messrs. Emery and Loring; of feedwater heaters; of fresh water for make-up purposes, whether carried in reserve tanks or supplied by evaporators, which were a development of this period; of grease extractors, to prevent the deposit of oil on the heating surfaces of boilers, accompanied by a great reduction in the quantity of oil used for internal lubrication.

Other developments were the use of metallic packing for piston rods and valve stems, of white metal for bearing surfaces, and the almost universal substitution of piston for flat slide valves for high- and intermediate-pressure cylinders, and for low-pressure cylinders of large diameter. Electric lighting was generally introduced, mechanical refrigeration had been applied to some extent, and artificial ventilation was introduced in naval vessels and passenger ships.

A few quadruple-expansion engines were built during the latter part of the period covered by this brief review, the most conspicuous examples being the twin engines of the *St. Louis* and *St. Paul*, built by the Cramp Company, and of the *Grand Duchess*, by Newport News, and a few Lake steamers of much lower power.

The general practice in the merchant marine was to operate all pumps necessary for propulsion from beams worked from one of the engine crossheads, but the Navy had early made use of independent feed and bilge pumps, and later of independent air and circulating pumps; and so satisfactory had been that experience that the merchant service was not slow to follow this practice for medium- and high-powered ships, except in respect to air pumps. For low-powered ships, air and bilge pumps are still operated from beams.

Boilers of merchant ships during this time were practically all of the cylindrical, or "Scotch," type. During the latter part of the period, the Navy showed a decided tendency to adopt water-tube boilers, its first torpedo boat, the *Cushing*, built by the Herreshoff Company having been equipped with Thornycroft boilers. Later, the monitor *Monterey* was provided with water-tube boilers, she being the first vessel of any navy larger than a torpedo boat to be so equipped. This action was taken as a result of the recommendation of Commodore Melville, the engineer-in-chief of the

Navy, to the Secretary of the Navy, that water-tube boilers be used for the major portion of her power as the readiest means of effecting a reduction in weight of machinery. This resulted in a competitive test of three types of boilers and a decision to use the Ward coil boilers for 75 per cent of her power. The ship had no difficulty in meeting the contract requirements, but the general adoption of water-tube boilers, except for torpedo boats, was not determined upon until some years later. In torpedo boats or destroyers, the pressure ran as high as 300 pounds with bent-tube boilers, and when water-tube boilers were adopted for battleships, to 265 pounds with straight-tube boilers. The effect of this great increase in pressure was to demand the production of solid cold-drawn steel tubes, which our tube mills were able to produce. A few Lake steamers were equipped with Babcock & Wilcox boilers, but, except in yachts, water-tube boilers found little favor outside the Navy. The first battleships to have them were the *Maine*, the *Missouri*, and the *Ohio*—the *Maine* with Niclausse, and the others with Thornycroft boilers—all in 1898. They were followed in 1899 by the *Georgia*, the *Nebraska*, and the *Virginia*—the *Georgia* and *Virginia* with Niclausse, and the *Nebraska* with Babcock & Wilcox.

For military reasons, as well as for constructional considerations, all naval vessels except those of very low power were provided with twin screws, but these found little application in the merchant service until high-powered ships such as the *St. Louis* and the *St. Paul* were built in 1895 for the transatlantic trade.

From 1890 to 1900 Congress made liberal appropriations for the Navy, especially after the Spanish War, some of the notable vessels completed during this time being the triple-screw cruisers *Columbia* and *Minneapolis*, whose speed of about 23 knots excited wide interest, being exceptional at that time—1893-1894—for vessels of their size. It was not until 1890 that a battleship was authorized, the Act for that year providing for the *Indiana* and *Massachusetts*, built by the Cramp Company, and the *Oregon*, by the Union Iron Works, which were the only shipyards equipped to build battleships. In 1895 the Newport News Company entered the field after completing three gunboats, and the Bath Iron Works and Moran Brothers Company, Seattle, in 1899. Bath had previously built several torpedo boats, two gunboats, and a monitor, and the Moran Company two torpedo boats, and while the jump from a torpedo boat to a battleship is a big one, it was successfully accomplished.

Other firms than those mentioned that contributed to the building of the Fleet were: Fore River Engine Company, destroyers and torpedo boats; Fore River Ship and Engine Company, a sheathed cruiser; George Lawley & Son, torpedo boats; Charles L. Seabury Company, gunboats and torpedo boats; Crescent Shipyard, a gunboat, a sheathed cruiser, a monitor and torpedo boats; Neafey & Levy, a sheathed cruiser and three destroyers; Harlan & Hollingsworth Company, destroyers and torpedo boats; Columbian Iron Works, a gunboat and torpedo boats; Maryland Steel Company, destroyers; Dialogue & Son, a gunboat; W. R. Trigg Company, machinery for the *Texas*, a sheathed cruiser, destroyers, and torpedo boats.

INTRODUCTION OF TURBINE DRIVE IN NAVAL VESSELS.

The foregoing covers the period from 1880 to 1900, but the succeeding thirty years witnessed still greater progress in marine engineering, and are specially noteworthy for the use of turbines, of oil fuel, and of mechanical and electric reduction gears between the turbine and the propeller shaft. The triple-expansion engine continued to hold its place as the favored propulsive agent except in cases where the great increase in power demanded

for increase in size of ship and superior speed forced the retirement of this reliable motor. During this entire period the power of merchant-ship engines did not greatly exceed 10,000 horsepower, as exemplified in such twin-screw ships as the *Finland* and *Kroonland*, built by the Cramp Company in 1902, the *Manchuria* and *Mongolia*, by the New York Shipbuilding Company in 1904, and the *Minnesota* and *Dakota* in the same year by the Eastern Shipbuilding Company, of New London, Connecticut. The two latter had Niclausse water-tube boilers, but their installation was not a success.

When the contract for the battleship *Delaware* was let to the Newport News Company in 1907, a similar one was entered into with the Fore River Company for the *North Dakota*, which was to be equipped with Curtiss turbines. It was realized that improvement would have to be made in reciprocating engines if they were to hold their own in competition with turbines, and wide departures were therefore made in the engines of the *Delaware* in an effort to accomplish this purpose. Cylinder ratios in previous engines had been 7 to 1, but with such engines it was necessary to admit live steam to the I.P. and L.P. receivers when running at full power. This was fatal to economy, and threw a disproportionate load upon the I.P. and L.P. cylinders, with attendant engine troubles. In the *Delaware*, the ratio was increased to 8 to 1; steam ports, instead of having the conventional curved outline, were made as straight and as direct as possible in order to reduce clearance; steam was superheated, and forced lubrication applied to the engines. This resulted in a notable increase in economy, greatly improved the operation of the engines, and was largely responsible for a slight increase in speed over that of the turbine competitor. Gratifying as was this performance, it was recognized that the limit had been reached with the reciprocating engine, and that if the size and speed of ships were to continue to increase, dependence would have to be placed on turbines.

Prior to this competitive test, another had been inaugurated in the building of three scout cruisers of identical hull construction but with different machinery installations. One had triple-expansion engines, another twin-screw Curtis reduction-gear turbines, both with identical boilers, and the third with 4-shaft Parsons turbines, with cruising turbines, but with a slightly different type of small-tube boilers from the other two. All passed the official trials successfully, the Parsons ship excelling and the reciprocating ship being last at full power. In service, however, the latter proved more reliable and more economical than the turbine-engined ships at cruising speeds, the order being reciprocating, Parsons, Curtis.

Following this experience, the next battleships contracted for were turbine-driven, through four shafts, Parsons turbines being used in both ships, as also in the two which followed a year later. This progressive move was not without its drawbacks, as the straight drive necessitated a higher speed of propeller than was conducive to propulsive efficiency, even when a slight sacrifice of economy of turbine was made in order to improve propulsive efficiency. It will not, therefore, be surprising to be told that the 1910 battleships had triple-expansion engines similar in all respects to those of the *Delaware*, which were giving remarkably good results in service. In the ships of the following year, the *Nevada* and *Oklahoma*, Curtis turbines were installed in the one and triple-expansion engines in the other. The turbines in this instance were a marked improvement over those of the *North Dakota*, and, with the addition of a geared cruising unit, showed great improvement in economy. The following year, 1912, marked the introduction of geared cruising turbines in all turbine-driven ships.

The first marine turbine built in this country for a seagoing merchant vessel was constructed by the Fletcher Company, Hoboken, N. J., for the

steamer *Governor Cobb* for the Eastern Steamship Company—the hull was built at Chester. A similar vessel was built in 1906 by the Bath Iron Works. Both had 3-shaft Parsons turbines of 4000 horsepower. The *Yale* and the *Harvard* followed a year later with Parsons turbines of 10,000 horsepower, also by the Fletcher Company. At this time there was built by the Cramp Company for the Southern Pacific Company the twin-screw steamer *Creole*, equipped with Curtis turbines of 8000 horsepower, operating in connection with Babcock & Wilcox boilers under natural draft, but the turbines proved a failure and were soon removed. In 1914 the same firm built the twin-screw steamers *Great Northern* and *Northern Pacific*, of 15,000 horsepower, with Mosher watertube boilers and 3-shaft Parsons turbines, and in 1928 the twin-screw steamer *Malolo*, with Parsons turbines of 25,000 horsepower and steam furnished by B. & W. boilers. In 1917 the Union Iron Works built the twin-screw steamer *Maui*, of 15,000 horsepower, for the American-Hawaiian line, with geared Parsons turbines and B. & W. boilers. It will be apparent from this that where the development of considerable power is necessary, the merchant service follows the lead of the the Navy in using turbines and water-tube boilers.

After the completion of the torpedo boats and destroyers of the 1898 program, all vessels of that class were equipped with turbines, some with 3-shaft Parsons, and others with 2-shaft Curtis, or Zoelly. The large number built during the World War were provided with twin-screw reduction-gear Parsons, Curtis, General Electric-Curtis, and Westinghouse turbines, and with Thornycroft, Yarrow, Normand, and White-Forster boilers. Many of the vessels built for the Shipping Board during the war were also turbine driven and equipped with water-tube boilers.

In looking for an engine that would satisfy the conditions of power, weight, and space for the 112 patrol vessels built during the war by the Ford Motor Company, the Bureau of Engineering decided upon the installation of a Poole geared turbine of 2500 horsepower, which is believed to be the first time this turbine was ever installed in a seagoing vessel.

Besides the major purposes to which turbines were put during the last thirty years, they have been extensively used for auxiliary purposes, the most important of which has been in driving forced-draft fans. The speed at which it was necessary to run fans driven by reciprocating engines resulted in so much trouble, and was the source of so many breakdowns, that in 1905 the Navy had recourse to electrically operated fans in battleships; but this was of short duration, for turbine-driven fans in destroyers had meanwhile proved so satisfactory that they soon were adopted as standard for battleships also.

OIL FUEL AND SUPERHEATED STEAM.

The first application of superheated steam in the Navy was in the battleships *Michigan* and *South Carolina*, which were laid down in 1905. The boilers were of the B. & W. type and had sufficient surface to give from 50 to 100 degrees of superheat. The results were so satisfactory that it was applied in all subsequent capital ships. Its use in the merchant marine was confined almost exclusively to some of the vessels built under the direction of the Shipping Board during the war. Its first application was probably in 1905, in the Lake steamer *James C. Wallace*, which had a triple-expansion engine of about 2000 horsepower and B. & W. boilers of 250 pounds pressure, with 50 degrees of superheat.

The use of oil fuel has done more than anything else to make possible the high sustained speed of our capital ships and destroyers, and of the

high-powered ships of the merchant marine. Its first application for marine use on a comparatively large scale was in 1903, when the *Mariposa*, of 2500 I.H.P., engaged in trade between San Francisco and Tahiti, was fitted to burn oil. In that case atomization was accomplished through the use of compressed air, which proved satisfactory except for the noise inseparable from compressed-air installations. That the proponents of the plan were not convinced of its reliability may be judged by the fact that provision was made to substitute steam for air in case of necessity. About the same time, the New York Shipbuilding Company equipped several ships of the American-Hawaiian line with oil-burning equipment on the Lassoe-Lovekin system, in which atomization was accomplished by low-pressure air supplied by a Roots blower. Meanwhile the Navy had been conducting exhaustive tests of all oil burners that were presented to the Liquid Fuel Board, whose report, in 1904, is a valuable compendium of all that was known about oil burning up to that time, and so impressed the Bureau of Steam Engineering with the importance of developing a satisfactory system that in 1906 oil was introduced in the *Delaware* as an auxiliary to the coal-burning equipment; but it was not until 1912 that oil was specified as the fuel for exclusive use in the battleships *Oklahoma* and *Nevada*, these being the first of such vessels to be so equipped. Atomization was accomplished by mechanical means, with provision for heating the oil to increase its fluidity so that it might be more easily handled by the service pumps. Earlier experience in the destroyers of the 1907 to 1911 programs, about 40 in number, had, however, been of such a satisfactory character as to give reasonable assurance that a return to coal would not be necessary. The merchant service was not slow to recognize the value of oil, not only as a substitute for coal, but also on account of the smaller boiler-room force required, and the reduced cost of maintenance of boilers so equipped. Some noteworthy installations are those of the *Leviathan*, *Great Northern*, *Northern Pacific*, *Matsonia*, *Maui*, *Malolo*, *Siboney*, *California*, *Virginia*, and *Pennsylvania*, to be presently referred to.

ELECTRIC DRIVE FOR MARINE PROPULSION.

The next important advance was in the application of electric drive to marine propulsion. In substance, the arrangement is merely the installation of one or more turbo-electric generators, the current from which operates one or more motors attached to the propeller shafts, and enables the turbines to be operated at a speed suitable for the best economy, and the propellers at speeds appropriate to propulsive efficiency. As the turbines always run in one direction, backing turbines, which were a fruitful source of trouble in all-turbine installations, are dispensed with. The introduction of electric drive came about through the offer of the General Electric Company to supply on a "no cure, no pay" basis the equipment for the collier *Jupiter*, of 7000 horsepower, the installation to be removed if it should prove unsuitable for the service on which the ship was engaged or unsatisfactory in operation. It not only measured up to every requirement of service, but possessed so many advantages from a military point of view that, in 1914, the Navy Department had no hesitation in installing it in the battleship *New Mexico*, and a year later in the *California* and *Tennessee*, the former with General Electric and the latter with Westinghouse equipment. The battleships of the 1916 program had similar installations, two by the General Electric and one by Westinghouse. All of these were from 27,000 to 30,000 horsepower. As the *Jupiter* had continued to give the greatest satisfaction, it was determined in 1917 to put electric drive of 60,000 horsepower

in the battleships of that program which were scrapped under the terms of the Washington Conference for the limitation of armaments, and in the six battle cruisers of 180,000 horsepower, four of which were also scrapped. The two remaining ones have since been completed as airplane carriers, and are now in service with the Battle Fleet, the fastest capital ships of any Navy. The Coast Guard has also installed electric drive in a number of their cutters, these being the first marine equipments with synchronous motors. The Panama-Pacific Line was so impressed with the success of electric drive in the Navy that they contracted with the Newport News Company for the construction of the twin-screw steamer *California*, with electric drive of 17,000 horsepower, for service between New York and San Francisco via the Panama Canal. She was completed in 1927, and has proved so satisfactory in service that the owners have equipped two sister ships, the *Virginia* and the *Pennsylvania*, with machinery of the same type.

INTRODUCTION OF DIESEL POWER.

The progress made in the development of the Diesel engine in this country has been much less than that made abroad. Our earliest Diesels were of very low power—about 120 horsepower—of the 4-cycle, single-acting type, and were installed in submarines, where they were satisfactory. As the size and surface speed of the submarine increased, troubles came thick and fast, especially with some of the 2-cycle engines, and it was only during the World War, and subsequent to our entrance, that fairly satisfactory engines were secured. Our extensive building program, authorized in 1916, was instrumental in obtaining engines that might be said to be satisfactory, but they were of only 600 to 1000 horsepower, installed two in a boat. Since then six fleet submarines have been authorized with twin propelling engines of 2000 to 2300 horsepower each. In 1911 a submarine tender was authorized. She was equipped with a 2-cycle engine of 1100 horsepower, which was anything but a success. In 1912, a fuel ship, the *Maumee*, was equipped with twin-screw 2-cycle engines of 2600 horsepower each, but they gave no end of trouble, chiefly on account of valve-scavenging difficulties.

A number of Diesel installations have been made in tugs and other small craft of the merchant marine, but comparatively few of much power in sea-going vessels. In recent years the Shipping Board has made the greatest contribution to the advancement of Diesel engineering in this country by awarding contracts for engines of from 2700 to 3300 horsepower to replace triple-expansion engines in twelve vessels constructed by that organization. The purpose was to determine the reliability of Diesels and their economy, and to further the development of that type of engine in this country. The engines were of the 4-cycle and 2-cycle types, single- and double-acting, and were built by the Worthington Company, the Electric Boat Company, Hooven, Owens, Rentschler, Busch-Sulzer Company, and the McIntosh & Seymour Company. Most of the vessels in which they were installed were employed on long routes, and the Shipping Board was so well pleased with the results that similar engines are to be installed in eight other vessels.

Gas engines, so extensively used in motorboats and small yachts, played an important part in keeping the submarine down during the war, and for that reason alone are entitled to a little more than passing notice. When it was decided to build a large number of submarine chasers, with special equipment for the detection of submarines, it was necessary that both the hulls and the machinery should be built where their construction would not offer the slightest interference to the construction of destroyers. As many

as 450 of these little vessels were built on the Atlantic, the Pacific, the Lake, and the Gulf coasts. A speed of 18 knots was contemplated in their design, and it was necessary to have engines of 600 horsepower to accomplish this. For many reasons it was desirable that this be in two units, but investigation developed the fact that there was not on the market a 300-horsepower engine that had demonstrated its reliability, and that could be put in quantity production with the slightest assurance that it could meet the dates of readiness of the hulls for its installation. It was known, however, that the Standard Motor Construction Company, of Jersey City, had supplied a large number of 220-horsepower engines for a similar but smaller boat for the British Navy, and inquiry developed the fact that these engines had been quite satisfactory under very trying service conditions. Decision was reached to install three of these in our chasers, and to accept the complication of three engines for the sake of reliability, a sine qua non in these boats, which of necessity would be manned by a small and an inexperienced crew. The decision was fully justified by the splendid service that these boats rendered.

No recital of the progress of marine engineering would be complete without mention of the work done in 1917 in repairing by electric welding the engines of the German ships interned in our ports, especially the transatlantic liners, which later were to render such splendid service in transporting troops to France. Upward of one hundred vessels had been seriously disabled, and to such an extent as to satisfy the perpetrators of the sabotage that the damage could not be repaired. They were confirmed in this opinion by a board of marine engineers convened under the direction of the Shipping Board, which reported that renewal of broken parts was the only remedy. Acceptance of this verdict would have made it impossible for us to use these vessels at all. Another board, composed of engineers of the Erie and New York Central Railroads, who had had experience in the use of electric welding in the repair of iron castings, were convinced that all castings that had been fractured could be repaired by welding. Their judgment was accepted and the work carried out under the direction of the engineer officer of the New York Navy Yard. On the Atlantic coast the damage consisted chiefly of broken cylinders, cylinder heads, valve chests, steam nozzles, and throttle and other steam valves. The repair was so successfully accomplished that this superb fleet of ships was soon ready to transport troops to the battlefields of France. Damage to vessel interned on the Pacific coast was confined chiefly to dry-firing boilers, with but little injury to the engines.—"Mechanical Engineering," April, 1930.

GEARED DIESEL MACHINERY.

It would perhaps be difficult to find a better example of what is frequently referred to as the ultra-conservatism of marine engineers than the fact that, although it is something like 10 years since the first geared Diesel installation went to sea, the principle of interposing mechanical-reduction gears between an internal-combustion engine and a ship's propeller is still regarded as something in the nature of a compromise, which should be avoided if at all practicable. There are, of course, exceptions to every rule; and the foregoing statement should be qualified by adding that in Germany—where the marine geared Diesel engine had its birth—engineers are generally disposed to regard this form of drive as possessing certain definite advantages in special cases. Those who have criticized marine engineers on the score of conservatism have frequently pointed to the more progressive methods of power production and transmission in vogue ashore. The demands of

shore and sea service, however, are so widely different in character that any such comparisons are often neither fair nor logical. It would be illogical, for example, to suggest that because the reciprocating internal-combustion engine of the motorcar has for a generation or more transmitted its power with almost absolute success through mechanical-reduction gears, the geared Diesel drive for sea-going ships should prove to be equally successful by proceeding along the same lines.

There are to-day quite a number of successful marine geared Diesel installations in operation; but this satisfactory position has not been attained without overcoming many complex problems, and it appears quite safe to say that the available land experience with geared internal-combustion engine drive made no material contribution to the solution of these problems. The principle of geared Diesel propulsion—as has been the case with many other important principles—was first applied owing to force of circumstances, and more or less as a compromise. At the close of the war, Germany found herself in possession of a large number of high-speed submarine Diesel engines for which there was no available commercial market; and as a quick, cheap and ready means of engining the first units of her cargo fleet then being reconstructed, a number of these engines were installed in merchant vessels.

The first sea-going installation of this kind was made by the Hamburg-Amerika Line in a twin-screw cargo ship of the *Havelland* * class trading to the Far East. This pioneer installation comprised two ten-cylinder submarine Diesel engines, each driving one of the twin screws through single-reduction mechanical gearing with a speed-reduction ratio of rather less than 3:1. The high-speed engine shaft was extended aft for about 20 feet through a flexible hollow shaft, on which the gear pinion was mounted. This formed, in effect, a variation of the so-called "Nodal" drive, then much in vogue, as a means towards damping torsional oscillations in double-reduction geared-turbine installations.

With the memory of several outstanding turbine reduction-gearing failures still fresh in mind, there were many experienced engineers at that time who held the view that mechanical-reduction gears would never endure the unven torque characteristics of the Diesel engine. These fears, however, were proved to be groundless as the performance of the *Havelland* gears, as well as that of the gears in several other Hapag cargo motorships similarly equipped about that time, was characterized by complete immunity from mechanical trouble of any kind. A logical development which followed the complete success of these early pioneer cargo-motorship installations was the design and construction of heavier-type high-speed Diesel engines for the propulsion of higher-powered motorships of the liner type, as exemplified by the group of vessels of the *Monte Sarmiento* † class, of which the Hamburg-South American Line took delivery soon afterwards.

Interest in the future of the geared Diesel drive has been considerably stimulated by the important order which the Hamburg-South American Line recently placed with Messrs. Blohm & Voss for two twin-screw 14,000-ton, geared-drive, motor-driven passenger liners of the *Monte Sarmiento* type, one of which is intended to replace the *Monte Cervantes*, unfortunately lost some little time ago. This is perhaps the most significant tribute which could be paid to the geared Diesel drive, as this company have had extended experience with this class of propulsion from 1924 onwards in three large twin-screw liners engaged on the long South American run.

* No. 136, Vol. XXV, p. 811.

† No. 178, Vol. XXXII, p. 44; and No. 178, Vol. XXXII, p. 416.

In their first geared Diesel ship—the *Monte Sarmiento*, delivered in 1924—the propelling machinery comprised four M.A.N. single-acting, four-stroke cycle, six-cylinder engines arranged in pairs, each pair driving a single propeller through single-reduction gears with a speed ratio of 2.8:1. The "Nodal"-drive flexible-shaft arrangement was originally fitted in this vessel; but when the basin trials were run, alarming vibrations and heavy chattering in the gear teeth took place, causing the bending of the flexible shafts and the breaking of the bearing bolts of the gears. As each pair of engines was coupled to a common gear-wheel and was fitted with a common starting device, the difficulty of ensuring that each engine would pick up at the same instant on the starting air apparently caused one engine to work against the other, and this phenomenon gave rise to torsional oscillations and resonance. Investigations showed that these oscillations were intensified by the spring effect of the flexible shafts, and accordingly the "Nodal" drive was abandoned in favor of rigid connection between the engines and gears, the only damping effect being that derived from the heavy flywheel. Torsion-graphs have shown that, except for slight variation in starting, the torque in the rigidly-connected geared drive is comparatively regular; and the builders—Messrs. Blohm & Voss, who have devoted extended research to the problems associated with the starting and running of Diesel engines geared in pairs on a common wheel—have no hesitation in guaranteeing absolute dependability with this system. After six years' continuous service, the original pinions and gears of the *Monte Sarmiento* show virtually no wear, and the owners' superintendent engineer has expressed his conviction that the gears will easily outlast the vessel.

The most important application of the geared Diesel drive so far adopted for marine work is to be found in the two twin-screw 17,000-ton transatlantic liners *Milwaukee* and *St. Louis*, of the Hamburg-Amerika Line. In these important vessels, the same form of drive as in the *Monte Sarmiento* class of ships has been adopted, *i.e.*, four engines arranged in pairs to drive twin screws. The machinery arrangement in the Hapag vessels, however, represents several important departures from that in the *Monte Sarmiento* class. In the first place, the prime movers are M.A.N. double-acting, two-stroke cycle, six-cylinder engines running at 230 R.P.M., which is reduced to 110 R.P.M. at the propellers; and whereas the aggregate power in the *Monte Sarmiento* class is only 6,000 B.H.P., each of the Hapag ships' machinery develops 12,500 B.H.P. A further important variation has been introduced; as, while the *Milwaukee*, constructed by Messrs. Blohm & Voss, has been fitted with rigidly-coupled pinions and gears, the *St. Louis*, built by the Bremer Vulkan Company, has Vulcan hydraulic couplings between the engines and the gears.

It is claimed—and apparently with some justification—that the damping effect of the Vulcan coupling relieves the mechanical gearing from any element of torque irregularity which might ordinarily arise from the engine, and, conversely, the coupling also provides a damping effect against shocks arising from the propeller. The respective merits of the rigidly and the hydraulically-coupled gears in these two important ships should be clearly demonstrated by a comparison of the operating results. Meanwhile, both vessels have completed their first transatlantic season with conspicuous success and immunity from mechanical trouble, and it can hardly be gainsaid that the all-the-year-round transatlantic run offers a very severe test for this comparatively new type of drive.

Reviewing the present position of the geared Diesel drive, as represented by these and other important applications to marine propulsion, it is apparent that there has been no suggestion of mechanical trouble, and this relia-

bility aspect is of primary importance in marine work. Other advantages attendant upon the development of power on board ship by a number of relatively small high-speed geared units are the greater potential freedom from vibration and the low head-room required. The latter consideration is obviously not unimportant in a passenger liner, as the arrangement of the decks overhead is greatly facilitated.

On the score of specific weight, the geared Diesel drive also shows to advantage, and this is apparent from a comparison of the weight characteristics of the machinery of the two Hapag vessels and of that of the normal direct-drive Diesel. A fair representative specific weight for a normal direct-drive Diesel engine to-day—making due allowance for the wide variety of arrangements available—is probably not less than 250 pounds per B.H.P.; whereas the published weight of the Hapag geared installation is 110 pounds per B.H.P. exclusive of scavenging pumps and gearing, or 125 pounds per B.H.P. with these included. For the development of large aggregate powers in vessels of the liner type, the facility with which a number of comparatively handy-sized engines can be coupled with the geared drive is in marked contrast with the position represented by the heavy, lofty, direct-drive engine. The excessive weight of the reciprocating masses of the latter gives rise to vibrational problems which are not always easy of solution, and the overhaul and adjustment of the working parts are correspondingly involved.

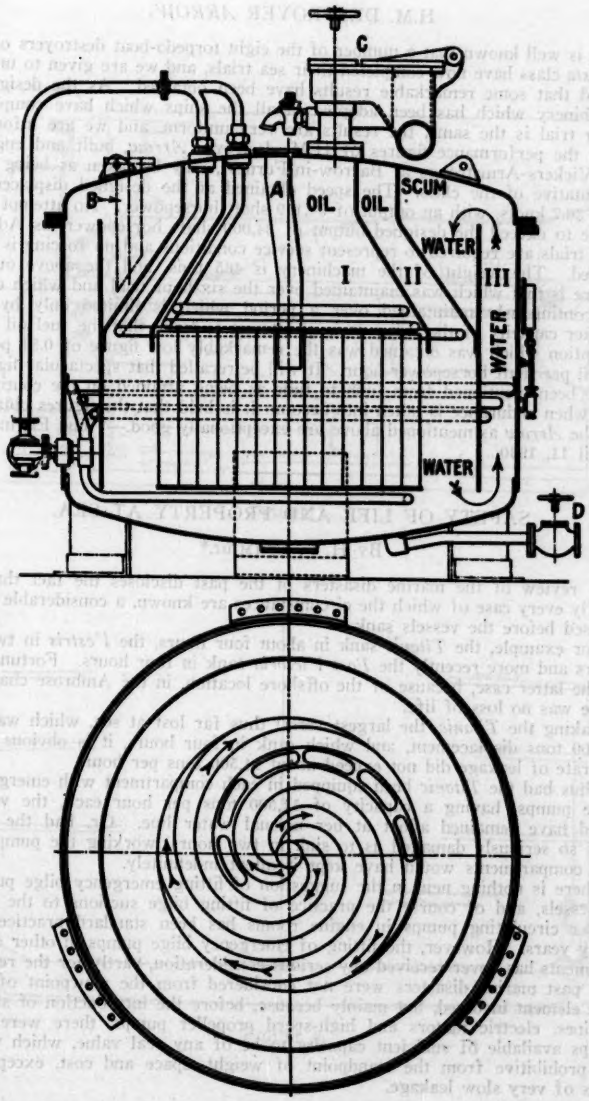
The geared Diesel installation for the German "pocket battleship" *Ersatz Preussen*, of 50,000 B.H.P. on four shafts—while admittedly a freak installation judged by mercantile standards—is indicative of the possibilities of the system. No system of propulsion, however, is wholly free from defects, and one of the disadvantages occasionally attributed to the geared Diesel drive is the objectionable noise which emanates from the gears. This problem of noise has arisen, it is true, in the Hapag *St. Louis*, but has been largely solved by careful insulation of the engine casings, so that the sound no longer penetrates to the passenger accommodation. The majority of mechanical gears have their own characteristic note, the emission of which it is difficult to eliminate in a metallic structure such as a ship; but the geared motor vessel is not alone in this respect. Not a few geared turbine-driven passenger liners might be equally criticized on this score; and even in the *Bremen*—the new holder of the Blue Riband of the Atlantic—the high-pitched hum of one of her port-side reduction gears is clearly audible on the main promenade deck. The high degree of precision now possible in gear-cutting, and the greater attention which is being devoted to insulation as a means of eliminating ship noises, are together factors which may be depended upon to remove this almost solitary, but by no means major, objection to the use of the geared Diesel drive.

In view of the prevailing opinion that the upper limit in what may, for want of a better term, be described as the "manageable" size of the high-powered, direct-drive Diesel engine has now been reached, and having regard to the apparent reluctance of marine engineers in general to adopt Diesel-electric propulsion, there would appear to be a wide potential field open for the association of mechanical-reduction gears with relatively small, high-speed Diesel engines where large aggregate powers at reasonably low weight are desired. By such a combination, the admittedly attractive economic advantages of the internal-combustion engine should, and could, be secured for the propulsion of all but the very largest and fastest passenger liners. For the latter, there does not as yet appear to be any serious rival to the high-pressure water-tube boiler, working in conjunction with the geared steam turbine.—"The Shipbuilder," April, 1930.

NEW TYPE OF OILY WATER SEPARATOR.

Apart from the necessity imposed by law of separating oil from bilge or ballast water when a vessel is in harbor or is approaching land, it is always an economical procedure to install a separator, since the oil thus obtained can be returned to the tanks. A number of designs of separator for dealing with oily bilge and ballast water have been introduced in recent years, but many of these suffer from the disadvantage that in order to obtain the non-turbulent flow which is essential to the efficient separation of the oil from the water, they occupy a large amount of space relatively to their capacity. This disadvantage has been cleverly overcome in the Comyn oil separator by the use of an arrangement of spirals so designed that they kill the turbulent flow of the mixture after it has entered the container with the minimum of eddying.

The Comyn oil separator, which is marketed by Separators Limited, 55, West Regent Street, Glasgow, and manufactured in a number of standard sizes by Alexander Stephen & Sons Ltd., Linthouse, consists of a cylindrical tank containing a set of multiple spirals and a pair of concentric cylinders forming two annular water passages round the spirals, the arrangement being as shown in the drawing reproduced above. The mixture of oil and water enters the separator through the pipe A, shown in the arrangement drawing, and passing tranquilly round the spirals, the bulk of the oil rises to the top in the space I, where it is withdrawn through the oil discharge valve C. When the water reaches the space beyond the spirals the remainder of the oil rises in the space II, whence it can be withdrawn through a connection (not shown in the drawing), on the oil discharge valve C, the water passing under the cylindrical baffle into the space III. From here it is withdrawn through the pipe B, shown dotted, any oil or scum accumulating at the top of this space being withdrawn through another connection to the valve box C, test cocks being provided to indicate when any oil has collected under the respective valves. A low-level test cock is also provided to show when a given depth of oil has accumulated above the spirals. Heating coils are fitted as shown to assist separation when necessary, and also to enable the separator to be cleaned in a few minutes at any time without opening up. Any sludge which collects can be drained out thorugh the valve D. It will be noticed that the valve C is fitted with an observation box, the whole being so arranged that no overflowing or flooding of the separator can take place. Further, a relief valve is fitted to prevent any excess pressure accumulating in the separator. It is claimed that this separator, apart from the advantage of occupying a small space, can be placed either above or below the water line, in any convenient space available. It is low in first cost, and having no moving parts, and being easy to clean, it is simple to operate, while its efficiency is very high. As exemplifying the efficiency of the Comyn oil separator, it may be mentioned that while two ships were recently undergoing survey in dock on the Clyde, the various tanks, the cofferdams, and all bilges in each ship were pumped completely empty through one of the standard Comyn oil separators, 6 feet in diameter by 5 feet high (total headroom required for dismantling being 7 feet 6 inches), and no difficulty was experienced in recovering the whole of the oil, amounting in each case to from 30 to 50 tons. The water, in all well over 1000 tons, was discharged direct into the dock under the observation of the harbor authorities. The recovered oil in each case was immediately used for generating steam in an oil-fired boiler with complete absence of spluttering or fluctuation of flame, the whole of the work being carried out to the entire satisfaction of the superintendent engineer concerned.—"Shipbuilding and Shipping Record," January 9, 1930.



General Arrangement of the Comyn Oily Water Separator.

H.M. DESTROYER *ARROW*.

It is well known that a number of the eight torpedo-boat destroyers of the *Acasta* class have now completed their sea trials, and we are given to understand that some remarkable results have been obtained. As the design of machinery which has been adopted in all the ships which have completed their trial is the same, the results are very uniform, and we are informed that the performance figures of H.M. destroyer *Arrow*, built and engined by Vickers-Armstrongs at Barrow-in-Furness, may be taken as being representative of the class. The speed obtained at the designed displacement was 36.7 knots, with an output of 34,119 shaft horsepower. No attempt was made to exceed the designed output of 34,000 shaft horsepower, as Admiralty trials are required to represent service conditions and no forcing is permitted. The weight of the machinery is 465 tons, and the above output figure is that which was maintained over the six-hour trial and which could be continuously maintained over a period which is limited only by the bunker capacity of the ship. It is of interest to learn that the fuel oil consumption which was obtained was the remarkably low figure of 0.81 pound of oil per shaft horsepower hour. It will be recalled that spectacular figures have been mentioned from time to time as being attained on the continent, but when endurance is taken into account it is held that the figures obtained on the *Arrow* as mentioned above are exceptionally good.—"The Engineer," April 11, 1930.

SAFETY OF LIFE AND PROPERTY AT SEA.

By H. F. Schmidt.*

A review of the marine disasters of the past discloses the fact that in nearly every case of which the circumstances are known, a considerable time elapsed before the vessels sank.

For example, the *Titanic* sank in about four hours, the *Vestris* in twelve hours and more recently the *Fort Victoria* sank in four hours. Fortunately in the latter case, because of the offshore location, in the Ambrose channel, there was no loss of life.

Taking the *Titanic*, the largest vessel thus far lost at sea, which was of 46,000 tons displacement, and which sank in four hours, it is obvious that the rate of leakage did not exceed about 11,500 tons per hour.

Thus had the *Titanic* been equipped in each compartment with emergency bilge pumps, having a capacity of 11,500 tons per hour each, the vessel could have remained afloat at her normal water line. Or, had the ship been so seriously damaged as to sink in two hours, working the pumps in two compartments would have kept it afloat indefinitely.

There is nothing new in the suggestion of fitting emergency bilge pumps in vessels, and of course the practice of fitting bilge suctions to the condenser circulating pumps in engine rooms has been standard practice for many years. However, the fitting of emergency bilge pumps in other compartments has never received any serious consideration, partly for the reason that past marine disasters were not considered from the viewpoint of the time element involved, but mainly because, before the introduction of steam turbines, electric motors and high-speed propeller pumps, there were not pumps available of sufficient capacity to be of any real value, which were not prohibitive from the standpoint of weight, space and cost, except in cases of very slow leakage.

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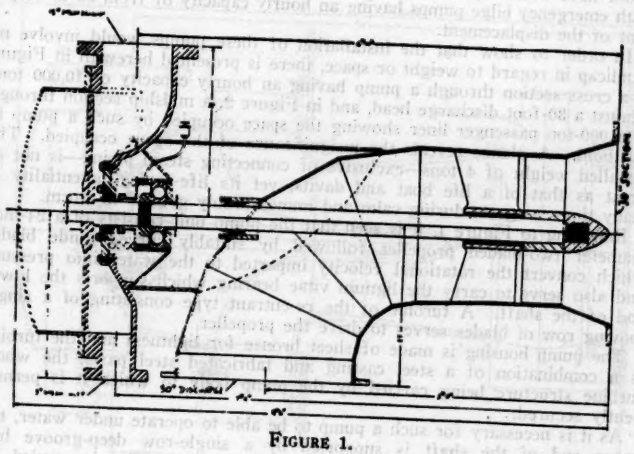

FIGURE 1.

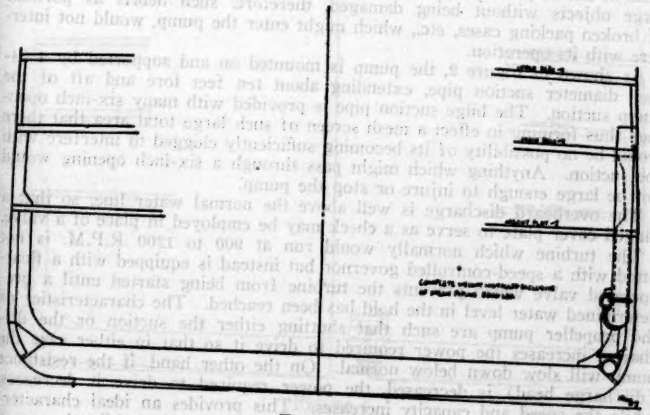

FIGURE 2.

From the record of past sea disasters, the writer believes that ninety per cent of all the vessels which have foundered or have been sunk in collision could have been kept afloat had they been equipped in each compartment with emergency bilge pumps having an hourly capacity of from 20 to 30 per cent of the displacement.

In order to show that the installation of these pumps would involve no handicap in regard to weight or space, there is presented herewith in Figure 1, a cross section through a pump having an hourly capacity of 10,000 tons against a 30-foot discharge head, and in Figure 2, a midship section through a 30,000-ton passenger liner showing the space occupied by such a pump in the hold. A glance reveals the insignificance of the space occupied. The installed weight of 4 tons—exclusive of connecting steam piping—is not as great as that of a life boat and davits, yet its life-saving potentiality is many times as great during calm and immeasurably greater in storm.

Referring to Figure 1, it is seen that the pump unit consists of a 24-inch diameter two-bladed propeller followed by suitably curved guide blades which convert the rotational velocity imparted to the water into pressure and also serve to carry the lignum vitae bearing which supports the lower end of the shaft. A turbine of the re-entrant type consisting of a single moving row of blades serves to drive the propeller.

The pump housing is made of sheet bronze for lightness and the turbine is a combination of a steel casting and fabricated steel parts, the whole turbine structure being carried by the pump body to which it is permanently secured.

As it is necessary for such a pump to be able to operate under water, the upper end of the shaft is supported by a single-row deep-groove ball bearing packed with grease. The thrust of the propeller is carried by the ball bearing which is designed with a very liberal factor of safety.

Because of its design, the two-bladed propeller, having a projected area ratio of but 60 per cent, is ideal for such service as it can pass comparatively large objects without being damaged, therefore, such debris as portions of broken packing cases, etc., which might enter the pump, would not interfere with its operation.

As shown in Figure 2, the pump is mounted on and supported by a 24-inch diameter suction pipe, extending about ten feet fore and aft of the pump suction. The bilge suction pipe is provided with many six-inch openings thus forming in effect a mesh screen of such large total area that there would be no possibility of its becoming sufficiently clogged to interfere with the suction. Anything which might pass through a six-inch opening would not be large enough to injure or stop the pump.

The overboard discharge is well above the normal water line, so that a hinged cover plate to serve as a check may be employed in place of a valve.

The turbine which normally would run at 900 to 1200 R.P.M. is not fitted with a speed-controlled governor but instead is equipped with a float-operated valve which prevents the turbine from being started until a predetermined water level in the hold has been reached. The characteristics of the propeller pump are such that shutting either the suction or the discharge, increases the power required to drive it so that in either case the pump will slow down below normal. On the other hand, if the resistance (discharge head) is decreased, the power required to drive it decreases, while the speed and capacity increases. This provides an ideal characteristic for an emergency bilge pump since, for example, while the pump shown in Figure 1 has a designed capacity of 10,000 tons per hour against 30-foot head at 950 R.P.M. if the leakage exceeded 10,000 tons per hour, the water would rise in the hold and thus reduce the head against which

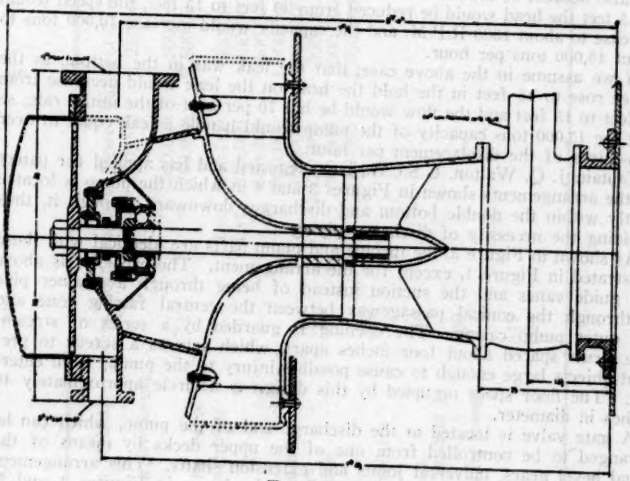

FIGURE 3.

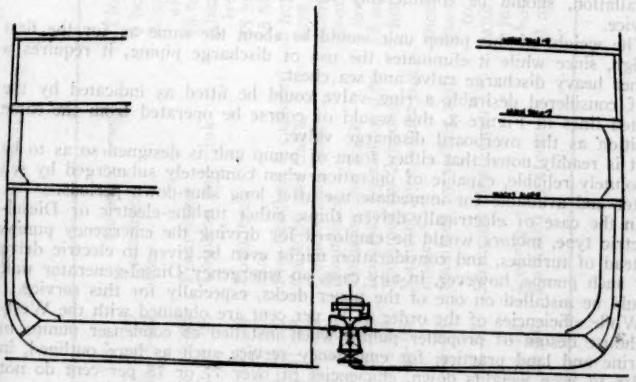

FIGURE 4.

the pump has to work. Consequently, the speed would increase and with it the capacity which increases not only because of the increase in speed but also because of the reduced resistance. Thus if the water rose to a height of 15 feet the head would be reduced from 30 feet to 15 feet, the speed would increase to about 1300-R.P.M. and the capacity would increase 10,000 tons to about 15,000 tons per hour.

If we assume in the above case, that the leak was in the bottom, as the water rose to 15 feet in the hold the head on the leak would decrease from 30 feet to 15 feet and the flow would be but 70 per cent of the initial rate, so that the 15,000 tons capacity of the pump would handle a leak equal to over two-thirds of the displacement per hour.

Captain J. Q. Walton, U.S.C.G., has suggested and has applied for patent on the arrangements shown in Figures 3 and 4 in which the pump is located partly within the double bottom and discharges downward through it, thus avoiding the necessity of discharge piping.

As shown in Figure 3, the turbine and pump parts are identical with those illustrated in Figure 1, except for the arrangement. The propeller is above the guide vanes and the suction instead of being through a strainer pipe is through the conical passageway between the central fairing cone and the outer pump casing. The opening is guarded by a series of streamline struts spaced about four inches apart, which acts as a screen to prevent objects large enough to cause possible injury to the pump, from entering. The floor space occupied by this design is a circle approximately 48 inches in diameter.

A gate valve is located at the discharge end of the pump, which can be arranged to be controlled from one of the upper decks by means of the usual bevel gears, universal joints and extension shafts. This arrangement would afford easier installation than the unit shown in Figures 1 and 2, inasmuch as it does not involve cutting through the decks and perhaps passenger spaces to install discharge pipes, also the time required for installation, should be considerably less in the case of ships already in service.

The weight of this pump unit would be about the same as for the first design, since while it eliminates the use of discharge piping, it requires a rather heavy discharge valve and sea chest.

If considered desirable a ring valve could be fitted as indicated by the dotted lines in Figure 3, this would of course be operated from the same position as the overboard discharge valve.

It is readily noted that either form of pump unit is designed so as to be absolutely reliable, capable of operation when completely submerged by sea water and available for immediate use after long shut-down periods.

In the case of electrically-driven ships, either turbine-electric or Diesel-electric type, motors would be employed for driving the emergency pumps instead of turbines, and consideration might even be given to electric drive for such pumps, however, in any case, an emergency Diesel-generator unit would be installed on one of the upper decks, especially for this service.

While efficiencies of the order of 85 per cent are obtained with the Westinghouse design of propeller pumps when installed as condenser pumps in marine and land practice, for emergency service such as here outlined, in order to keep weights down, efficiencies of over 77 or 78 per cent do not seem justified inasmuch as the higher efficiency affects not only the pump, but also the piping which must necessarily be large.

When one thinks of passenger ships with thousands of boiler horsepower available, going to sea with a large number of passengers and crew and

Vessel	Date	Cause of Loss	Time Taken to Sink	Lives Lost	Value of Vessel	Value of Cargo	Tonnage
Homestead	7–21–27	Foundered	6 hours	...	Est. 600,000	1,250,000	Est. 6,868
Cabo Hatteras	3– 5–27	Fire	Abandoned a few hours after fire	...	" 500,000	500,000	" 4,594
Vestris	11–12–28	Foundered	48 hours	112	" 700,000	2,500,000	" 10,494
Floridian	9– 1–28	Collision and sunk	Abandoned by crew in less than one hour	...	Ins. 340,000	800,000	6,765
Chief Maquilla	11–30–28	Foundered	48 hours	...	Est. 350,000	400,000	9,394
Kochican	5–10–25	Fire	15 hours	..	" 350,000	500,000	2,255
Galilee	8–26–26	Fire beached & capsized	5 hours	...	" 500,000	1,500,000	6,287
Kershaw	5–31–28	Collision and Sunk	Less than one hour	...	450,000	600,000	2,741
Fort Victoria	12–28–29	Collision and sunk	About 3 hours	...	1,250,000	300,000	7,784
Bomardier	2– 9–21	Foundered	About 48 hours	...	500,000	500,000	3,162
Craster Hall	6– –27	Collision and Sunk	4 hours	...	750,000	1,250,000	4,319
Ville Du Havre	10–23–73	Rammed by sailing ship	16 minutes	80			
Oregon	3–14–86	Rammed by Schooner	7 hours, 30 minutes	...			
Elbe	1–30–95	Rammed by steamer not known	Short time	359			
La Bourgogne	6– 4–98	Rammed by sailing ship	Questionable reports ½ to 2 hours	600			
Republic	1–23–09	Rammed by steamer	Sank while being towed into port	4			
Titanic	4–14–12	Iceberg	4 hours	1517			
Empress of Ireland	5–29–14	Rammed by steamer	10 minutes	1024			

SUNK IN COLLISION AND REPORTED DURING THE LAST FIFTY YEARS.

			Lost
1870	Jan.	24—American ship *Oneida*, off Yokohama	115
1871	Jan.	27—Steamer *Kensington* collides with bark *Templar* off Cape Hatteras. Both wrecked	150
1873	Jan.	22—British steamer *Northfleet*, off Dungeness	300
	Nov.	25—French Line steamer *Ville du Havre* by sailing ship *Loch Earn*. Sunk in 16 minutes	230
1875	Nov.	4—American steamer *Pacific*, off Cape Flattery	236
1878	Sept.	3—British steamer *Princess Alice*, in the Thames	700
1879	Nov.	7—American Steamship *Champion* by sailing ship *Lady Octavia*, 15 miles from Delaware Lt. Sh.	31
1880	June	11—American steamer *Narragansett*, Long Island Sound	27
	Nov.	24—French steamer *Uncle Joseph*, off Spezzia	250
1882	July	4—Steamer *Sciota*, in Ohio River	57
1884	April	18—Bark *Pomena* and steamer *State of Florida*, off Coast of Ireland. Both sunk	150
	July	22—Spanish steamer *Oigon* and British steamer *Lexham*, off Cape Finisterre. Both sunk	150
1886	Mar.	14—Cunard steamer *Oregon*, with unknown schooner 18 miles from Fire Island Light. Both sunk	none
1887	Jan.	20—Steamer *Kapunda* with bark *Ada Nelmore*, off Brazil	300
	Nov.	19—Steamer *W. A. Scholten*, English Channel	134
1888	Aug.	14—Steamship *Geiser*, by the *Tingvalla*	105
	Sept.	12—An Italian Steamship and the Steamship *La France* collided near Canary Islands	89
1890	Oct.	29—Steamship *Viscaya*, off Barnegat	70
1891	Mar.	17—Steamer *Utopia*, Anchor Line, sunk off Gibraltar	574
	Sept.	10—Italian steamer *Taormica* sunk in Mediterranean	50
1893	June	22—British battleship *Victoria* by *Camperdown*, off Tripoli	350
1895	Jan.	30—German steamer *Elbe* by British steamer *Crethie* in North Sea	335
1898	July	4—French line steamer *LaBourgogne* by British sailing ship *Cromartyshire*	560
1902	July	21—Steamer *Primue* by steamer *Hansa* in the Elbe	112
1903	June	7—French steamer *Libau* near Marseilles	150
1907	July	20—American steamers *Columbia* and *San Pedro* on the California coast	100
1908	Mar.	23—Japanese steamer *Matsu Maru* near *Makedate*	300
	April	25—British cruiser *Gladiator* by American Liner *St. Paul*, off Isle of Wight	30
1909	Jan.	23—White Star steamer *Republic* by steamer *Florida*, off Nantucket	6
	Nov.	14—Steamer *Scyne* by steamer *Onda*, off Singapore	100
1912	April	8—Nile steamer near Cairo, Egypt	200
	April	14—White Star steamer *Titanic* by iceberg	1517
1913	Mar.	5—German destroyer *S-178* by cruiser *Yorck*	66
1914	May	29—Canadian Pacific steamship *Empress of Ireland* sunk by Danish collier *Storstad* in St. Lawrence	1204

representing tremendous investments for vessels and cargos, it would seem imperative that immediate provision should be made to utilize this new and perfected means for preventing sinking of ships. Aside from the humanitarian viewpoints however, it would seem like only good business to make such provisions since the cost should soon be more than repaid through the reduction in insurance rates.

Accompanying this article are tables listing data concerning marine casualties of a nature which this propeller type emergency bilge pump is designed to prevent.—"Marine Journal," April 1, 1930.

BOOK REVIEWS.

THE NAVAL BLOCKADE, 1914-1918. By Louis Guichard, Lieutenant in the French Navy, attached to the Historical Section of the French Ministry of Marine. Translated and edited by Christopher R. Turner. Published by D. Appleton & Company, New York, N. Y. Price, $3.50.

Following a logical and easily-followed plan, this volume pictures the complicated development, and the long-delayed accomplishment of the allied plan for the "economic encirclement" of the Central Powers during the World War. The author has marshalled an imposing mass of data, drawn largely from the customs reports of all the nations engaged, to demonstrate what a powerful factor toward ultimate victory the gradual strangulation of German imports proved to be.

He outlines in great detail the consecutive steps taken by the Allies in building up the commercial blockade during the first three years of the war, and gives sound reason for their much-criticised vacillation by pointing out the effect of each pronouncement on the great neutrals, and on the allied powers themselves. He shows how the blockade became a reality during the last year and a half, with the entry of the United States as a belligerent, despite Germany's resort in retaliation to the tremendously effective weapon of unrestricted submarine warfare. He pays what must be a somewhat reluctant tribute to the German nation as he pictures their patriotic self-denial, their inventive genius in substitution, and their adaptability in meeting the hardships and deprivations brought on them by the economic blockade as the war progressed to its conclusion.

To anyone particularly interested in knowing why the French delegation to the Disarmament Conference in London is holding out for a naval tonnage of over 700,000, and why France demands the right to build up a great submarine force, this book is recom-

mended for careful reading. While the author demonstrated the failure of the German policy of unrestricted submarine warfare as it was applied against the allied naval blockade of 1914-1918, he shows clearly the reasons why it failed—reasons which might never control the outcome of submarine warfare applied by a nation differently situated than was Germany during that time.

ROGER W. PAINE,
LIEUTENANT COMMANDER, U.S.N.

SALEM VESSELS AND THEIR VOYAGES. VOL. IV.

The Essex Institute of Salem, that notable repository of maritime history, has added a fourth volume to this fascinating series. The skillful and sympathetic pen of George Granville Putnam has prepared this valuable material for us, and another book of audacious commercial adventure is available.

The guts and go of early peoples, more politely called the pioneer spirit, budded and bloomed in Salem. There was no sea too wide, no port too far, for a ship of these people to cross and cast anchor in their search for cargo. In getting further acquainted with the men and ships of this great early port, one learns more and more of those whose remarkable capabilities had a large share in laying and shaping the keel of this great and prosperous country.

This volume is principally devoted to the activities of the distinguished Rogers family, their ships and the men that sailed them. They boldly sent them pioneering to Arabia, to Madagascar, to the Fiji Islands and to Australia. Building up a trade with the natives of these countries was one of hazard as these records show. To trade with cannibals, before their consciences were cramped by the influence of the missionaries, required tact and fearlessness. It was not judicious for a person of succulent physique to place too much confidence in the good manners of the natives of the Fijis. Mr. Putnam indicates that a lean and unpromising appearance saved more than one man of Salem.

This book is inspiring reading to the last page, filling one with admiration for the insistent, indomitable, and courageous personalities that are described. Not only did they overcome the dan-

gers and problems of navigation, but they resisted the losses so frequently imposed by treachery, robbery, and murder. The shipmaster was of necessity forever on his guard. Once a vessel stranded on the shores of these countries, her weakened situation exposed her to pillage and her crew to massacre.

One of the principal articles of trade at the Fiji Islands, and developed by the seamen of Salem, was beche-de-mer, a sea animal gathered from the reefs and cured for the Chinese market. They learned to cure these animals from a gang of pirates and natives living on the island of Amboyna. From these dangerous acquaintances they also bought several sugar boilers with which to prepare the cargo. The boilers had belonged to a brig a short time before, the master and crew of which they had murdered and the vessel burned. One of the first voyages paid $60,000; these boilers prepared four cargoes, after which the natives carried on the work. Salem maintained this profitable trade for many years.

One of the first beche-de-mer masters was Benjamin Vanderford, who afterwards served as South Sea pilot for the Wilkes Exploring Expedition of the Navy, and with which he died.

The Navy always found a cooperating friend in Salem. Not only for the services of that distinguished secretary, Benjamin W. Crowninshield, or for the contribution of their frigate *Essex,* which has taken such a notable place in Naval annals, but also for the remarkable services of its privateers both in the Revolution and the War of 1812.

Salem was quick to be stirred by the *Chesapeake-Shannon* affair fought off that port, especially by the heroism and conduct of Lawrence. The leading men of the port fitted out a brig manned by master mariners, and sailed under a flag of truce for Halifax where they successfully solicited the body of Lawrence. Upon returning there was prepared a funeral of "imposing magnitude."

The short biography of Capt. William Driver, to select one of a number, is an epic of romance dealing with a bold resourceful seaman, a diplomatic trader, a vigorous American, and an ornament to the merchant marine.

Interspersed between engrossing incidents and short narratives are many interesting pictures of Salem ships, and shipmasters who looked like statesmen.

GERSHOM BRADFORD.

THE BALANCING OF OIL ENGINES. By W. Ker Wilson, Published by Charles Griffin and Company, Limited, 42 Drury Lane, London, W C 2, and J. B. Lippincott Company of Philadelphia.

The oil engine was developed from a mere idea at the end of last century to the commanding position it occupies today by adjusting it to the peculiarities of its application. Thus the speed of the modern marine oil engine, for instance, varies from 60 to 100 R.P.M. in merchant motorships, to 300 to 400 R.P.M. in submarine installations, to 600 to 1000 R.P.M. in recent cruisers and battleships, to 1000 to 1500 R.P.M. in motor boats and up to 2500 R.P.M. for aircraft; the power installed varies from 5 horsepower in motor boats to 50,000 horsepower in a battleship, while the number of cylinders varies from the proverbial "one-lunger" to as many as 16 cylinders on one shaft.

Every departure from the heretofore common ranges of speed, power and number of cylinders was fraught with dangers and since the effects of unbalance grow as the square of the speed it is evident that the balancing problem of the oil engine is one of the principal problems which must be given increased attention. Balancing of rotating and reciprocating masses is not a problem introduced by the advent of the oil engine, the high speed steam engine was responsible for a number of books on this general subject, but there are certain characteristics of this problem which are peculiar to the multi-cylinder oil engine and a book devoted to the treatment of the oil engine balancing is therefore a welcome addition to the shelf of any engineer who is either engaged in the design of such engines or is responsible for the selection or operation of oil engines. The author, who belongs to the engineering department of a well-known British oil engine builder, is eminently qualified to write the book, and the book therefore is not a mere congregation of mathematical formulae of a theorist but a readable treatment of the problem by a practical engineer. The author prefers the analytical method, devoting only one chapter to the exposition of the graphical method, which is recommended for checking only. One chapter is devoted to the problems peculiar to each number of cylinders from one to eight with a chapter for

engine foundations and another one for all other related problems of the oil engine. The book has 279 pages, is well illustrated by photographs and sketches of different types of engines built, and by many numerical solutions of the balancing problem of actual engines.

E. C. MAGDEBURGER.

THE STRENGTH OF SHAFTS IN VIBRATION. By J. MORRIS, PUBLISHED BY CROSBY LOCKWOOD AND SON, STATIONERS' HALL COURT, LUDGATE HILL, LONDON, E. C. 4, PRICE 30 S. OD. 189 PAGES.

The title of this book suggests material of interest to a modern engineer, confronted as he is with many forms of shaft vibration. However, the author in his preface characterizes this book more aptly—"the main features of this work are the *elegant* general mathematical results." Presumably the elegance of the mathematical solutions of the abstract problems which the author developed out of the mass of problems presented by vibrating shafts would appeal to the mathematician but a practical engineer gathers out of a casual perusal of the book only the disdain of a theorist towards all things earthly, such as a broken shaft, stalled machinery, losses of time and money, etc. A more appropriate title to the book would appear to be "Mathematical Theory of Shaft Vibration." Not that the mathematical investigation is not important in itself, not that we do not need the cooperation of mathematicians, but the book is too mathematical to be engineering and therefore could not very well be considered of interest to a practical engineer whose curiosity it excites by its title.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

THE ANNUAL BANQUET.

The annual banquet of the Society was held at the Willard Hotel in Washington on the evening of May 9th. A record attendance of about two hundred and seventy members and guests, included leading representatives of the Houses of Congress, foreign attaches, the Naval and Military Services, the Coast Guard, and civilian engineering and shipbuilding organizations. For the first time in its history the Society was honored by the presence of a lady in the person of the Honorable Ruth Hanna McCormick, member of the Committee on Naval Affairs of the House of Representatives. Mrs. McCormick was introduced by the Toastmaster and enthusiastically received.

Following dinner a most interesting program of addresses was presented. Rear Admiral H. E. Yarnell presided and Rear Admiral S. S. Robison acted as toastmaster. Assistant Secretary of the Navy Ernest Lee Jahncke spoke in high terms of the Navy, particularly of the Engineering Branch which he believes should be more fully understood and appreciated. The subject of "Parity" was discussed by the Honorable Burton L. French with special reference to the recent London treaty. Mr. French declared that parity per se as between two nations is impossible of attainment due to the many and varied conditions which are involved. He believed that due regard must be given treaty obligations, that new construction should be carried on in a conservative way and that the types and characteristics of Naval ships should be determined by the experts in the Navy itself. Doctor Karl T. Compton discussed " The Activities of a Physicist" and recited some interesting personal experiences during the war. Doctor Compton is at present head of the Department of Physics at Princeton, and is to be President of Massachusetts Institute of Technology this fall. " The Navy and its relation to the Merchant Marine" was the

subject of an excellent address by Mr. Henry G. Smith, who is well known in the Navy, as well as in commercial fields, being a graduate of the Naval Academy, Class of '91. He spent several years in the Construction Corps, from which he resigned to enfer private shipbuilding. He is at present Vice President of the Bethlehem Shipbuilding Corporation, and President of the National Council of American Shipbuilders.

Mr. Henry F. Schmidt was presented with the medal (clasp) of the Society for the 1929 prize essay. Mr. Schmidt's paper, "Theoretical and Experimental Study of Condenser Scoops," was published in the February number of the JOURNAL. Mr. Schmidt enjoys with Rear Admiral Dyson the distinction of twice winning the prize essay award.

The banquet of 1930 will be remembered as one of the most enjoyable of these annual events.

JOINT DIESEL MEETING.

On May 8th, the Society collaborated with the Washington Society of Engineers and with the Washington Sections of the Society of Automotive Engineers and American Society of Mechanical Engineers in a joint Diesel Engine Meeting held in the auditorium of the Interior Department. A paper on "The Diesel Engine situation today" was presented by Mr. Max Rotter, Vice President of Busch-Sulzer Bros. Diesel Engine Company, "Diesel Engines for Submarines" by Mr. E. Nibbs, Chief Engineer of the Electric Boat Company, and, "Light weight Diesel Engines," by Mr. O. D. Treiber, President of the Treiber Diesel Engine Company. These papers will be printed in the August number of the JOURNAL.

MEMBERSHIP.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Davis, H. F. D., Commander, U.S.N.

Ducray, Julian A., Lieut. U.S.N.R., Bureau of Engineering, City Hall, San Francisco, Calif.

Lambert, John R., Electrician, U.S.N.

McGinitie, Marshall, Lieut., U.S.N.R., Washington Iron Works, 6th Ave., South and Atlantic St., Seattle, Wash.

Metzel, Jeffrey C., Lieutenant, U.S.N.

Osburg, Daniel, Machinist, U.S.N.

Tusler, Floyd A., Lieutenant (CC), U.S.N.

CIVIL.

Brown, J. J., Manager, Marine Service Dept., Westinghouse Electric and Mfg. Co., 460 W. 34th St., New York, N. Y.

Hale, H. C., 1514 Lexington Building, Baltimore, Md.

Hopkins, Stanley, Joint Managing Director, Kiesselbach Steam Accumulators, Ltd., London, England.

Kirgan, John, care Ingersoll-Rand Co., Phillipsburg, N. J.

Kyle, R. K., Vice President and Manager, Commercial Engineering Corporation 18th and E Sts., N. W., Washington, D. C.

McNulty, J. W., care Westinghouse Electric & Mfg. Co., Lester, Pa.

Maginnis, William, U. S. Shipping Board, 2063 Navy Building, Washington, D. C.

Raidabaugh, John A., 47 Richfield Road, Upper Darby, Pa.

Spear, Harry B., President, Hampton Roads Shipbuilding Corporation, Bankers Trust Building, Norfolk, Va.

Wood, C. E., Service Manager, Blood Bros. Co., 105 Dalla St., Allegan, Mich.

ASSOCIATE.

Carageorge, C. G., care Cox and Stevens, 521 5th Ave., New York, N. Y.

Collom, Samuel H., President, Pennsylvania Flexible Metallic Tubing Co., West Schoolhouse Lane, Germantown, Philadelphia, Pa.

Galvache, H. E. José, Royal Spanish Navy.

Googin, T. M., Westinghouse Electric & Mfg. Co., Lester, Pa.

Hasbrouck, L. P., Westinghouse Electric International Co., 150 Broadway, New York, N. Y.

Lidiak, J. P., 74 Berkley Avenue, Lansdowne, Pa.

Manera, Edmunda, Lieutenant, Argentine Navy.
Moyer, David W., 10 West 53d St., New York, N. Y.
Quiroga, F. A., Royal Spanish Navy.
Veiga, Victor, Lieutenant, Portuguese Navy.
Ward Horace, 18th and E Sts., N. W., Washington, D. C.
Watkins, William H., Y. M. C. A., Seattle, Wash.

NAVAL ENGINEERS PRIZE ESSAY AWARDS.

1897 W. W. White, Passed Asst. Engineer, U.S.N.
"Steam Consumption of the Main and Auxiliary Machinery of the U.S.S. *Minneapolis*."

1899 Prof. W. F. Durand—1st.
"Electric Propulsion of Torpedo Boats."
B. C. Ball—2nd.
"Indicator Diagrams of Multiple-Expansion Engines."

1900 W. R. Worthington, Lt. Comdr., U.S.N.
"Corrosion of Boiler Tubes in the U. S. Navy."

1901 John R. Edwards, Lt. Comdr., U.S.N.
"The Coaling of Warships at Sea."

1902 F. W. Bartlett, Lt. Comdr., U.S.N.
"The Repair Ship a Military Auxiliary to the Naval Fleet in Peace and War."

1903 Chas. W. Dyson, Lt. Comdr., U.S.N.
"Screw Propellers of U. S. Naval Vessels."

1904 Ernest N. Janson, M. E.
"Steam Turbines, with a Special Reference to their Adaptability to the Propulsion of Ships."

1909 Chas. W. Dyson, Comdr. U.S.N.
"Screw Propeller Criticism and Notes on Screw Propeller Design Based on Actual Standardization Trial Results of U. S. Vessels."

1926 S. E. Slocum
"Practical Application of Modern Hydrodynamics to Main Propulsion."

1927 Henry F. Schmidt
"Some Screw Propeller Experiments with Particular Reference to Pumps and Blowers."

1929 Henry F. Schmidt
"Theoretical and Experimental Study of Condenser Scoops."